NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

JPL - Technical Report No. 32-814

Generant Controller Development for the Advanced Liquid Propulsion System (ALPS)

W. F. MacGlashan, Jr.

JET PROPULSION LABORATORY
CALIFORNIA INSTITUTE OF TECHNOLOGY
PASADENA, CALIFORNIA

November 15, 1965

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

Technical Report No. 32-814

Generant Controller Development for the Advanced Liquid Propulsion System (ALPS)

W. F. MacGlashan, Jr.

Duane Dipprey

D. Dipprey, Manager
Liquid Propulsion Section

JET PROPULSION LABORATORY
CALIFORNIA INSTITUTE OF TECHNOLOGY
PASADENA, CALIFORNIA

November 15, 1965

Prepared Under Contract No. NAS 7-100
National Aeronautics & Space Administration

CONTENTS

CONTENTS (Cont'd)

TABLES

FIGURES

FIGURES (Cont'd)

ABSTRACT

10650

In the ALPS system (Advanced Liquid Propulsion System) the propellant tank is pressurized with gases generated by the decomposition of hydrazine in the gas generator. The function of the generant controller is to meter the flow of hydrazine to the gas generator so that constant pressure is maintained in the propellant tank during all firing periods regardless of variations in the rates at which propellants flow out of the tank.

The generant controller is essentially a remote-sensing, single-stage, spring-loaded regulator. Four controller versions were built and tested. Controller 4 incorporates the best features of the preceding three controllers. Results of water tests that simulate expected operating conditions are recorded.

Special features of these controllers, such as the diaphragm backup ring and the Belleville spring package which were developed as a result of this study, are discussed. The suitability of these special features for scalability and for other components is pointed out.

An Appendix is included which describes the function of the generant controller in the ALPS system and in the *Mariner* '66 system.

Author

I. INTRODUCTION

In the ALPS system, the propellant tank is pressurized with gases generated by decomposition of hydrazine in a gas generator (see Appendix). The purpose of the generant controller is to meter the flow of hydrazine to the gas generator so that constant pressure is maintained in the propellant tank during all firing periods regardless of whatever variation may occur in the rates at which the propellants flow out of the tank.

II. CONTROLLER FUNCTION

The ALPS generant controller is essentially a remote-sensing, single-stage, spring-loaded regulator. Each of the four controllers (designated as 1, 2, 3, and 4) covered in this report consists of three basic parts: a valve through which the generant flows, a pressure-sensing piston, and a spring assembly. Flow through the controller is varied by changing the annular gap between the knife edge and the seat in controllers 1, 2, and 3, and by changing the gap between a movable ball and a fixed circular seat in controller 4. Regulation of the flow is accomplished by the opposition of a preset reference force exerted by the spring assembly and the force exerted by the sensing pressure on the piston. The net difference between the two forces tends to open or close the flow gap from the equilibrium position. In other words, when the sensed pressure is low, the pressure force is less than the spring force, so the valve seat is opened; when the pressure force is high, it overcomes the spring force, so the valve seat is closed.

Delivered outlet pressure from the controller is not a significant design parameter in the ALPS application; since the controller is designed to be insensitive to it. Rather, propellant tank pressure, which the controller senses and to which it responds, should be maintained by the controller within very narrow limits.

III. CONTROLLER REQUIREMENTS

Flow. The current design is sized for a nominal generant flow rate of 0.1384 lb/sec at a pressure drop across the controller of 75 psi or less. The controller must be able to operate precisely and stably over a flow rate range of 20:1.

Pressure. Generant controller inlet and outlet pressure varies from a maximum of 1500 psi down to a minimum of 500 psi or less because of the "blowdown" principle used in the generant feed circuit (see the Appendix for a description of the blowdown principle). Sensing port or regulated pressure is to be set somewhere between 200 and 300 psi; whatever preset pressure is chosen, however, should be maintained by the controller within very narrow limits.

Leakage. External leakage of 300-psi nitrogen gas from the sensing mechanism must be kept below 10 cm^3/year (stp). External leakage of 1500-psi hydrazine from the liquid-containing part of the controller must be kept below 1 cm^3/year.

Seat leakage over the range of 0- to 1500-psi hydrazine shall not exceed 0.1 cm^3/hour.

Environment. The generant controller must perform reliably after storage periods of up to 1 year in a spacecraft environment. This means that the exterior of the controller has to be immune to high vacuum and radiation while the interior must be compatible with hydrazine for service and water for testing. Operating temperature of the ALPS system is 70±30°F so no severe temperature cycling problems will be faced unless the controller is used in a system which must be heat-sterilized. If sterilization is necessary, three cycles in which the temperature is raised from ambient to 300°F, held for 24 hours at that level, and lowered back to ambient, must be survived without damage or change in performance. It appeared that an all-metal design, such as versions 3 and 4 (Figs. 1 and 2), would be most likely to withstand these environmental conditions.

Lubrication. A thin film of Apiezon L grease is applied to the diaphragms and aluminum gaskets to facilitate retention during assembly and to assist in obtaining a leak-tight seal. External parts such as sliding fits, support annuli for backup rings and Belleville springs, bolt threads, etc. are lightly lubricated with Apiezon L either alone or mixed with Molykote Z. Apiezon L grease is compatible with and insoluble in hydrazine.

For flight, the grease could be omitted since toggle motion has replaced journal friction in controllers 2, 3, and 4, which therefore could be operated dry. Dry lube could be baked onto controller 1 at points of sliding contact.

Material Compatibility. Internally, the controller must be compatible with hydrazine and water. Externally, the controller must withstand the effects of hydrazine, water, and salt (NaCl) spray.

All three ports (inlet, outlet, and sensing) are in contact with the propellant (hydrazine).

Materials. The controllers are essentially of all-metal construction, although rubber discs to back up the diaphragms were used in the earlier controllers 1 and 2 (Figs. 3 and 4).

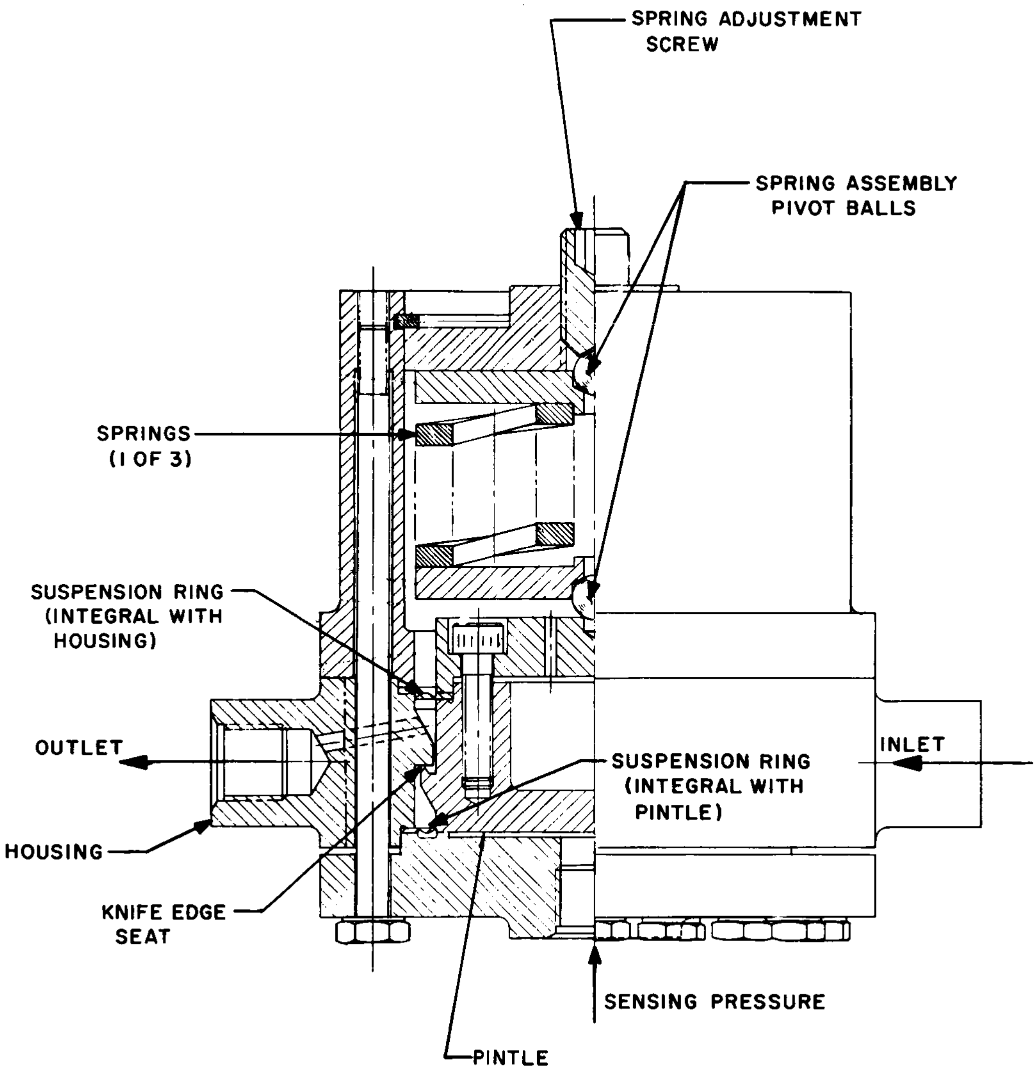

Fig. 1. Generant controller 3 cross section

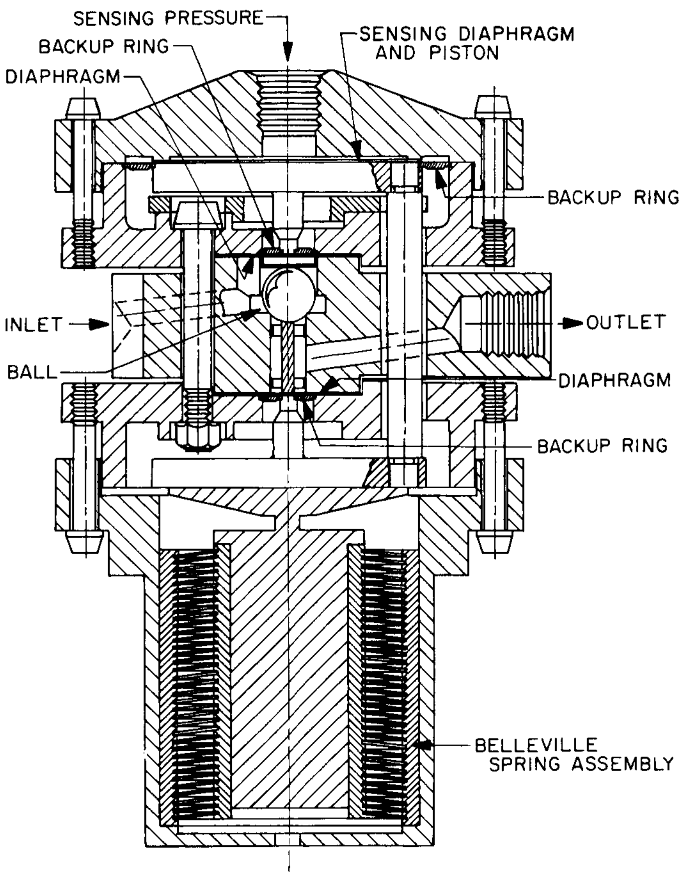

Fig. 2. Generant controller 4 cross section (ball pushed off seat)

Bolts are constructed of A-286 stainless steel or aluminum alloy and are retained by Ny-Lok inserts or by drilling the heads for lock wire.

Parts requiring high strength or hardness are machined from 17-4 PH or 17-7 PH CRES (corrosion-resistant steel) and heat-treated. Hard anodize was applied to some external parts of controller 1. Internal aluminum parts are left unanodized because hydrazine dissolves the aluminum oxide finish.

Aluminum parts in contact with hydrazine are made of 6061-T6 alloy. Gaskets are made of 1100-0 aluminum sheets.

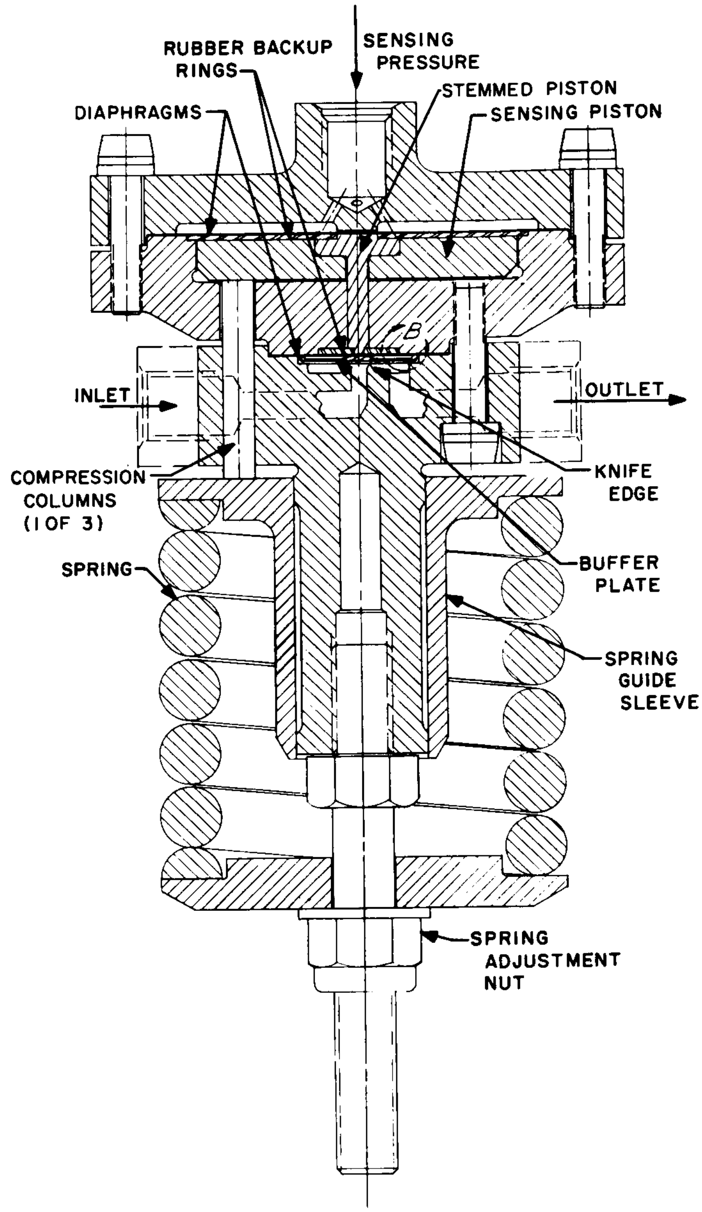

Fig. 3. Generant controller 1 cross section

Diaphragms are constructed of 1100-0 aluminum or 300 series CRES.

Backup rings and external structural parts are machined from 7075-T6 aluminum alloy.

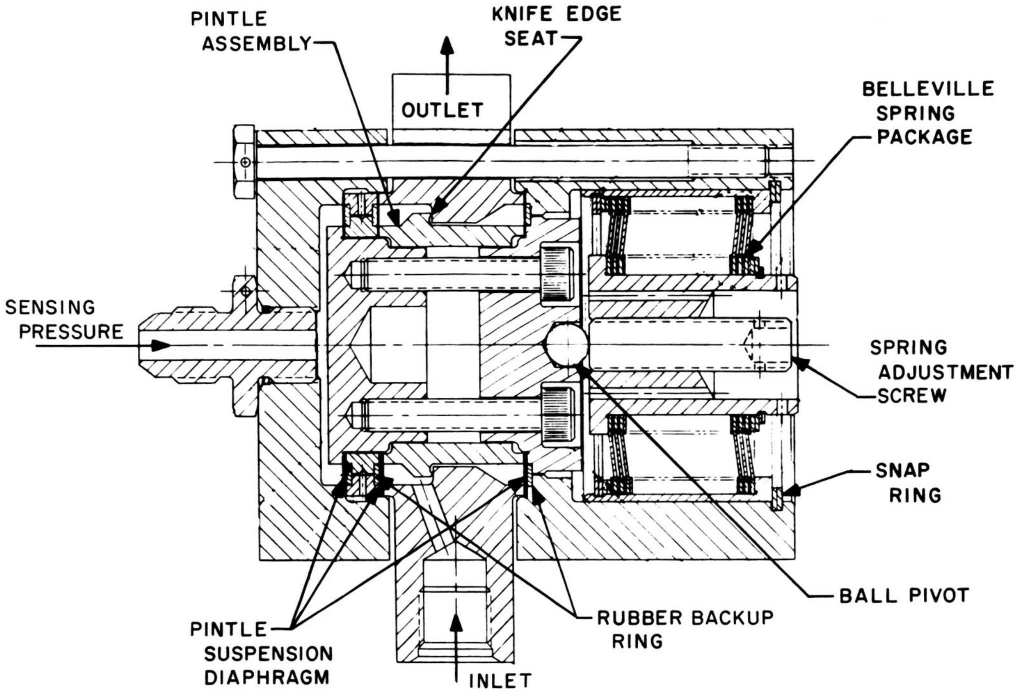

Fig. 4. Generant controller 2 cross section

IV. CONTROLLER 1

A. Description*

Figure 5 is an external view of controller 1. Figure 6 shows the parts and Fig. 3 is a cross-sectional drawing of controller 1.

The knife edge seat and the sensing port diaphragm-sealed piston arrangement are similar to those utilized in previous back pressure valve designs. For the generant controller application, a floating buffer plate of soft aluminum was added to engage the knife edge seat at shutoff, and a rubber ring was added between the diaphragm and the concentric reciprocating parts which the diaphragm seals.

The diaphragm-sealed sensing port piston is opposed by the spring. The spring cap is piloted on the housing body and is joined to the sensing piston by three compression columns which are slip fits in the housing body. The sensing piston, columns, and spring cap form a rigid reciprocating yoke. A floating stemmed piston, axially

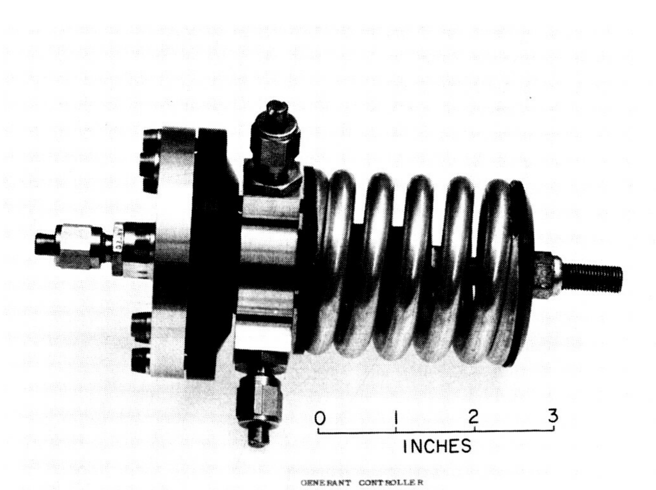

Fig. 5. Generant controller 1

*The general operation common to all four generant controllers is described in Section II. Table 1 gives drawing numbers, seat diameters and materials, etc., for each controller.

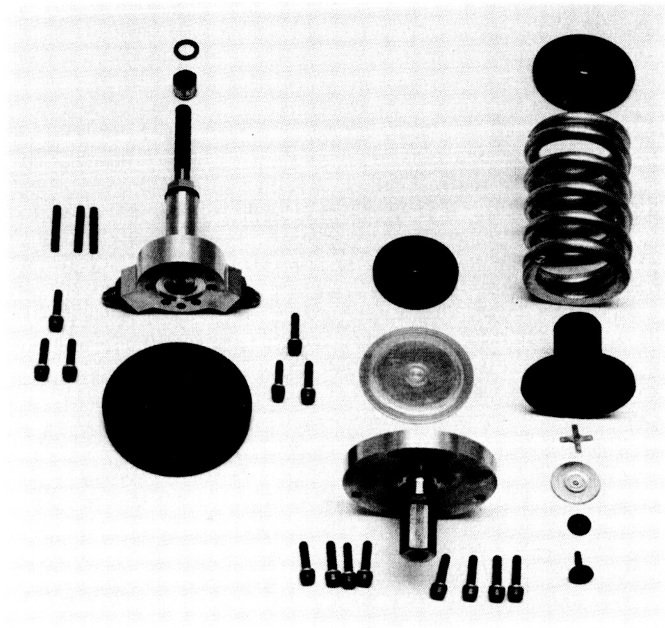

Fig. 6. Generant controller 1 exploded view

aligned with the seat, is flush with the face of the sensing piston and extends to the aluminum diaphragm that seals the internal seat cavity. The stemmed piston and the sensing piston move together as sensing pressure forces the yoke toward the seat knife edge until the stem pushes the buffer plate onto the knife edge to shut off flow. At shutoff, the stemmed piston has reached the limit of its downward travel, but the sensing piston can continue downward independently until equilibrium with the spring is reached. Thus the maximum closure force that can be transmitted to the knife edge is the product of the area of the OD of the stemmed piston and the unit sensing pressure. This dual piston arrangement prevents damage to the knife edge by limiting the closing force.

The buffer plate is keyed against rotation and contacts a peripheral rim just prior to flexing onto the knife edge. This assures proper alignment during seating.

A rubber ring is placed over the sensing piston and stemmed piston to cover the annular joints. Both pistons are recessed to receive the rubber ring so that the clamped sealing diaphragm will be flat. During yoke movement, the rubber backup ring smoothes out the abrupt annular level changes of the pistons which results in much greater diaphragm life because of the increased radius of diaphragm curvature.

In like manner, the internal seat cavity diaphragm is rubber-backed to extend diaphragm life. The diaphragm transmits the closing force and separates the stemmed piston from the buffer plate.

This controller has an inherent unbalance because of the 0.130-in.-diam set area, which amounts to about 5-psi variation in sensing pressure when either the inlet or outlet pressure varies 1000 psi. In addition, the sensing

Table 1. Physical comparison of generant controllers

Generant controller designation	JPL drawing No.	Sensing port effective diaphragm diam, in.	Weight, lb	Type of reference force spring	Spring force range, lb	Knife edge seat diam, in.	Knife edge material	Seat material
1	D 911 6051	2.00	2.90	1 helical compression	628–942	0.13	6061-T6 aluminum alloy	1100-0 aluminum
2	J 911 6170	1.50	1.90	1 Belleville unit	354–530	1.50	17-7 PH stainless steel	6061-T6 aluminum alloy, gold-plated
3	J 911 6173	2.00	2.70	3 helical compressions	628–942	2.00	17-7 PH stainless steel	6061-T6 aluminum alloy, gold-plated
4	J 911 6609 and Sketch No 65 X 03 300	2.00	2.90	2 Belleville units	628–942	0.250	(ball) ⅜ in. ceramic	17-4 PH stainless steel

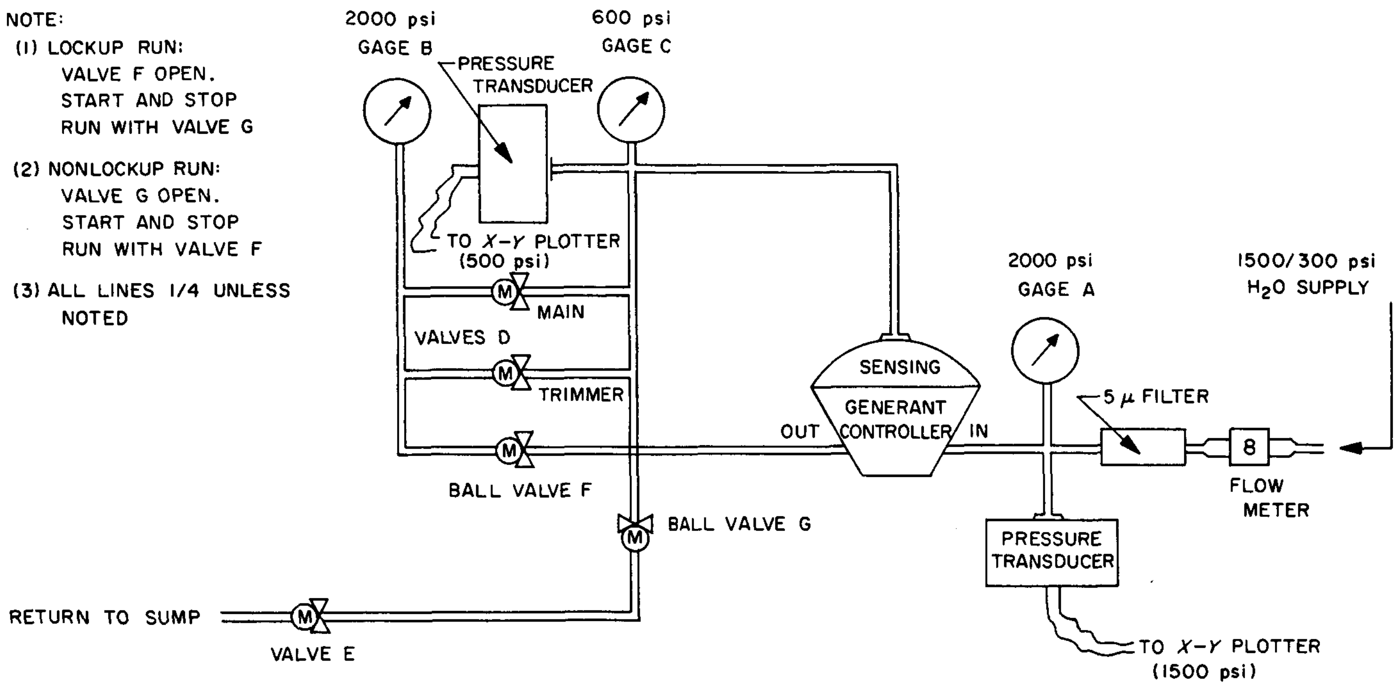

Fig. 7. Generant controller test setup for fixed flow rate and blowdown runs

pressure can vary about 3 psi due to spring rate during the 0.014-in. nominal seat travel. Under ideal conditions these two pressures tend to partially cancel each other. All rubbing surfaces are hard-anodized, however such friction as is present does have an effect on sensing pressure.

Table 2. Comparative controller regulation at 500-psi inlet pressure and variable flow rates

Flow rate, lb/sec of H_2O	Sensing pressure, psi			
	Controller 1	Controller 2	Controller 3	Controller 4
0.01 increasing	255	251	256	265
0.03	252	236	250	258
0.06	253	229	245	258
0.10	251	225	239	256
0.15	249	220	229	260
0.10 decreasing	254	231	239	260
0.06	256	233	245	264
0.03	253	238	250	267
0.01	257	248	256	267
Total psi deviation from nominal 255-psi sensing pressure	+2 −6	−4 −35	+1 −26	+1 +12
Note: These tests were performed using the test setup shown in Fig. 7.				

B. Performance

The controller was tested in the setup shown schematically in Fig. 7. The results of these tests are recorded in Tables 2–7. An explanation of the tables and comments on the controller characteristics are included in Section VIII.

Table 3. Comparative controller regulation at 1500-psi inlet pressure and variable flow rates

Flow rate, lb/sec of H_2O	Sensing pressure, psi			
	Controller 1	Controller 2	Controller 3	Controller 4
0.01 increasing	271	–	–	–
0.03	263	–	265	265
0.06	256	300	257	256
0.10	254	290	253	255
0.15	251	236	250	255
0.10 decreasing	254	291	253	266
0.06	261	–	255	266
0.03	263	–	260	258
0.01	275	–	284	–
Total psi deviation from nominal 255-psi sensing pressure	−4 +20	−19 +45	−2 +29	0 +11
Note: These tests were performed using the test setup shown in Fig. 7.				

Table 4. Effect of induced downstream ΔP on regulation at 500-psi inlet pressure at 0.15 lb/sec flow of H_2O

Additional psi pressure drop ΔP introduced at "out" port	Sensing pressure, psi			
	Controller 1	Controller 2	Controller 3	Controller 4
45 145	237 233	218 210	227 216	258 254
Total psi deviation from 255-psi nominal sensing pressure	−18 −22	−37 −45	−28 −39	+3 −1

Note: These tests were performed using the test setup shown in Fig. 7.

Table 5. Effect of induced downstream ΔP on regulation at 500-psi inlet pressure at 0.03 lb/sec flow of H_2O.

Additional psi pressure drop, ΔP introduced at "out" port	Sensing pressure, psi			
	Controller 1	Controller 2	Controller 3	Controller 4
45 145	254 256	257 241	251 249	258 259
Total psi deviation from 255-psi nominal sensing pressure	+1 −1	−16 +2	−6 −4	+4 +3

Note: These tests were performed using the test setup shown in Fig. 7.

Table 6. Effect of induced downstream ΔP on regulation at 1500-psi inlet pressure at 0.15 lb/sec flow of H_2O

Additional psi pressure drop ΔP introduced at "out" port	Sensing pressure, psi			
	Controller 1	Controller 2	Controller 3*	Controller 4
45 445 1145	252 259 246	241 230 205		256 257 253
Total psi deviation from the nominal 255-psi sensing pressure	−3 −6 −9	−14 −25 −50		+1 +2 −2

*Controller 3 failed before data could be obtained.
Note: These tests were performed using the test setup shown in Fig. 7.

Table 7. Effect of induced downstream ΔP on regulation at 1500-psi inlet pressure at 0.03 lb/sec flow of H_2O

Additional psi pressure drop ΔP introduced at "out" port	Sensing pressure, psi			
	Controller 1	Controller 2	Controller 3*	Controller 4
45 445 1145	263 261 257	300 300 252		– 266 266
Total psi deviation from the nominal 255-psi sensing pressure	+8 +6 +2	+45 +45 −3		– +11 +11

*Controller 3 failed before data could be obtained.
Note: These tests were performed using the test setup shown in Fig. 7.

V. CONTROLLER 2

A. Description*

Figure 8 is an external view of controller 2. Figure 9 shows the parts and Fig. 4 is a cross-sectional drawing of controller 2.

This controller utilizes a pressure-balanced knife edge seat. The pintle, which contains the knife edge, is made of heat-treated 17-7 PH CRES and is suspended on three 0.003-in.-thick, rubber-backed soft aluminum diaphragms. The rubber backing (1/32-in.-thick rubber washer) is to prolong diaphragm life by reducing creasing during flexure. The pintle is spring-opened (Belleville spring package) and is closed by sensing pressure. Both ends of the internal seat cavity and one end of the sensing port chamber are sealed by the pintle suspension diaphragms. The diaphragms are free to flex axially on an unclamped annulus of 1/8-in. radial width. Pressure balance (to assure that upstream or downstream pressure variations have no effect on pintle movement) is achieved by making the knife edge seat diameter identical to the effective diameter of the diaphragms. The knife edge seats against a flat gold-plated surface of the 6061-T6 aluminum alloy housing. A 0.0005-in. layer of nickel was applied to the aluminum surface by the Kanigen process prior to the application of a 0.001/0.002-in. layer of gold plating. The purpose of the gold plating is to provide a soft sealing surface to contact the knife edge so that shutoff can be obtained with minimum force. The space between the sensing chamber and the flow chamber diaphragms is vented to atmosphere. The Belleville spring package is adjusted by a central set screw that bears against a steel ball embedded in the knife edge assembly.

*The general operation common to all four generant controllers is described in Section II. Table 1 gives drawing numbers, seat diameters and materials, etc., for each controller.

B. Performance

The force to flex the diaphragms was higher than had been anticipated.

The controller was tested in the setup shown schematically in Fig. 7. The results of these tests are recorded in Tables 2–7. An explanation of the tables and comments on the controller characteristics are included in Section VIII.

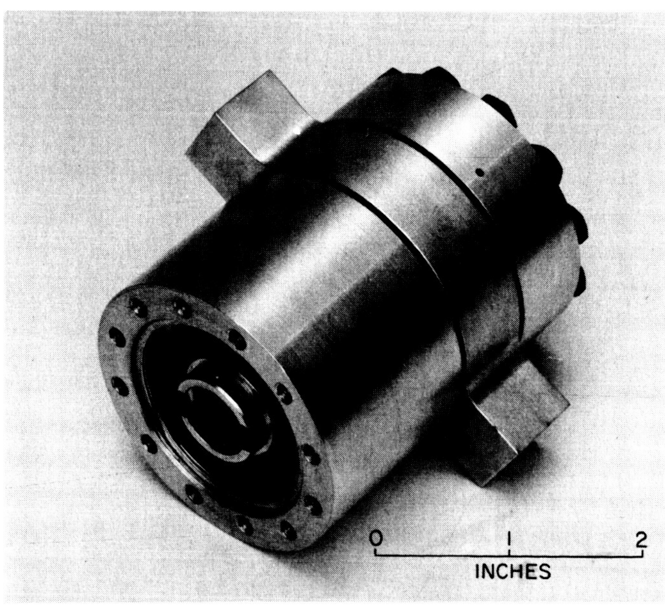

Fig. 8. Generant controller 2

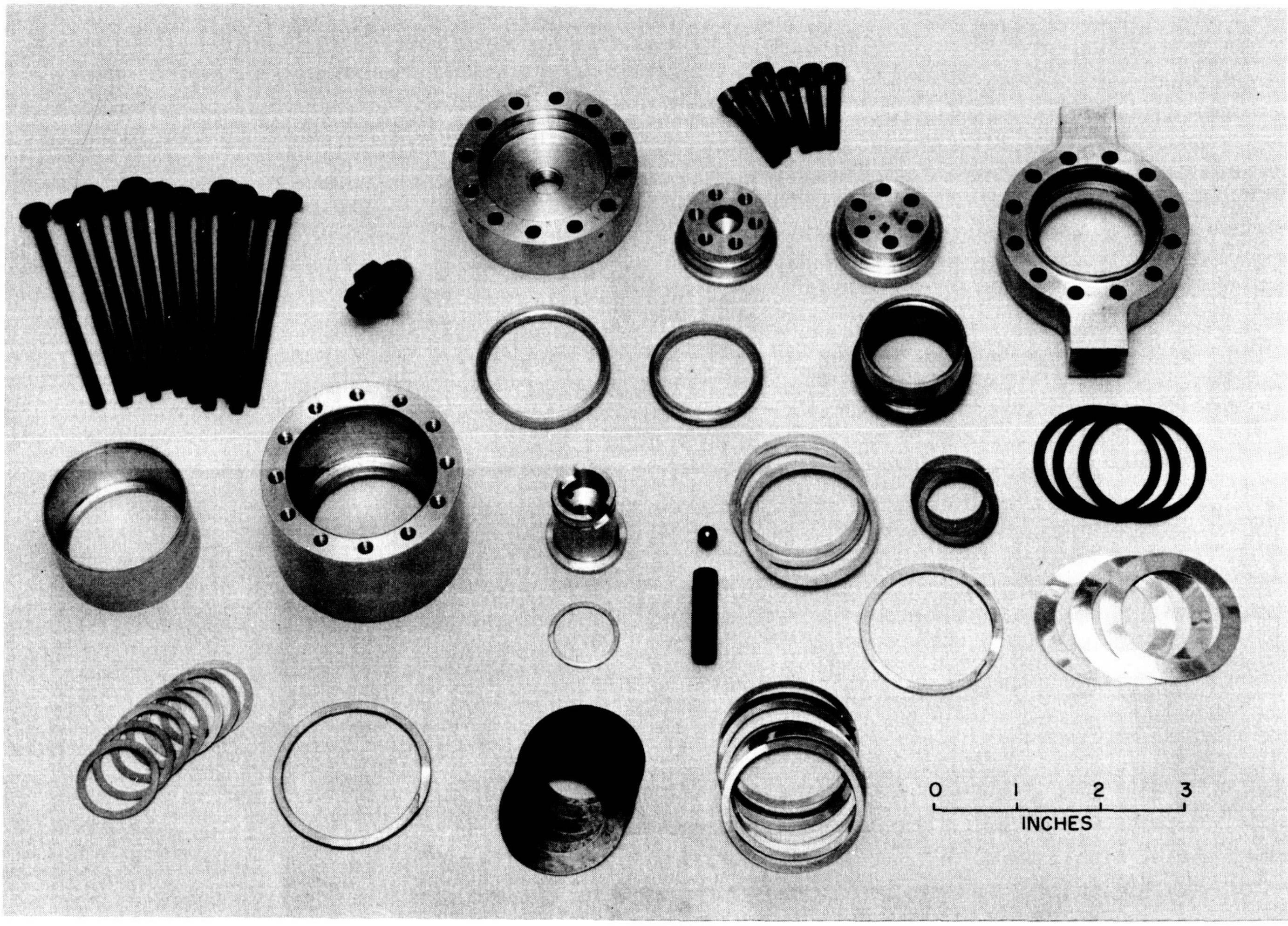

Fig. 9. Generant controller 2 parts

VI. CONTROLLER 3

A. Description*

Figure 10 is an external view of controller 3. Figure 11 shows the parts and Fig. 1 is a cross-sectional drawing of controller 3.

This is an all-metal, pressure-balanced controller with the knife edge seat. The pintle, which contains the knife edge, is made of heat-treated 17-7 PH CRES. The 6061-T6 aluminum alloy housing contains a flat gold-plated surface which engages the knife edge. The gold-plated seat is designed to assure tight shutoff with a minimum seating force. The pintle is suspended on two rings. One ring is integral with the pintle and the other ring is integral with the housing. The suspension rings are gold-plated to serve as gaskets when the unit is bolted together. The inner and outer peripheries of the rings are machined to be thin as possible and yet be able to safely sustain the proof pressure shear stress. The midpoint of the ring

*The general operation common to all four generant controllers is described in Section II. Table 1 gives drawing numbers, seat diameters and materials, etc., for each controller.

annulus is thickened to carry the bending load during proof testing. The spring rate of the suspension rings is very high but because of the large seat diameter, the required axial travel is very small. The ring, which is integral with the pintle, is a barrier between the sensing cavity pressure and the seat cavity pressure.

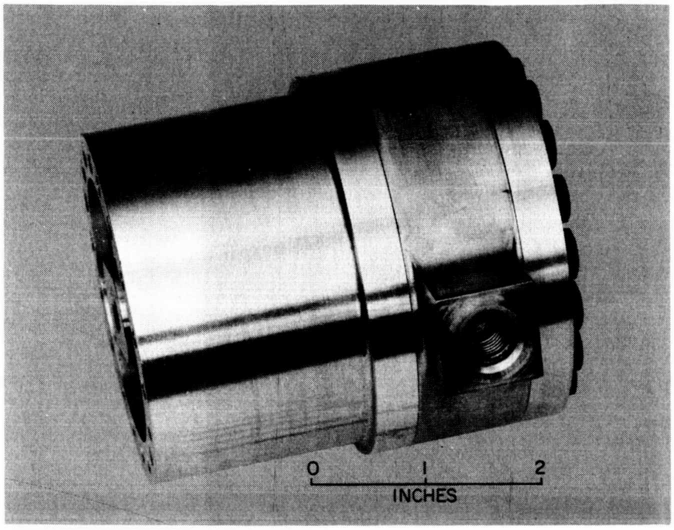

Fig. 10. Generant controller 3

Pressure balance is achieved by making the effective diameter of the suspension rings identical to that of the knife edge.

The pintle is spring-opened (cluster of three helical compression springs) and is closed by sensing pressure. The spring cluster is self-aligned with a steel ball at each end and is adjusted by means of a set screw in the spring housing cap.

B. Performance

The controller was tested in the set-up shown schematically in Fig. 7. The results of these tests are recorded in Tables 2–5. An explanation of the tables and comments on the controller characteristics are included in Section VIII.

Two units were tested. No data were obtained on the first unit because of damage to the seat by contamination

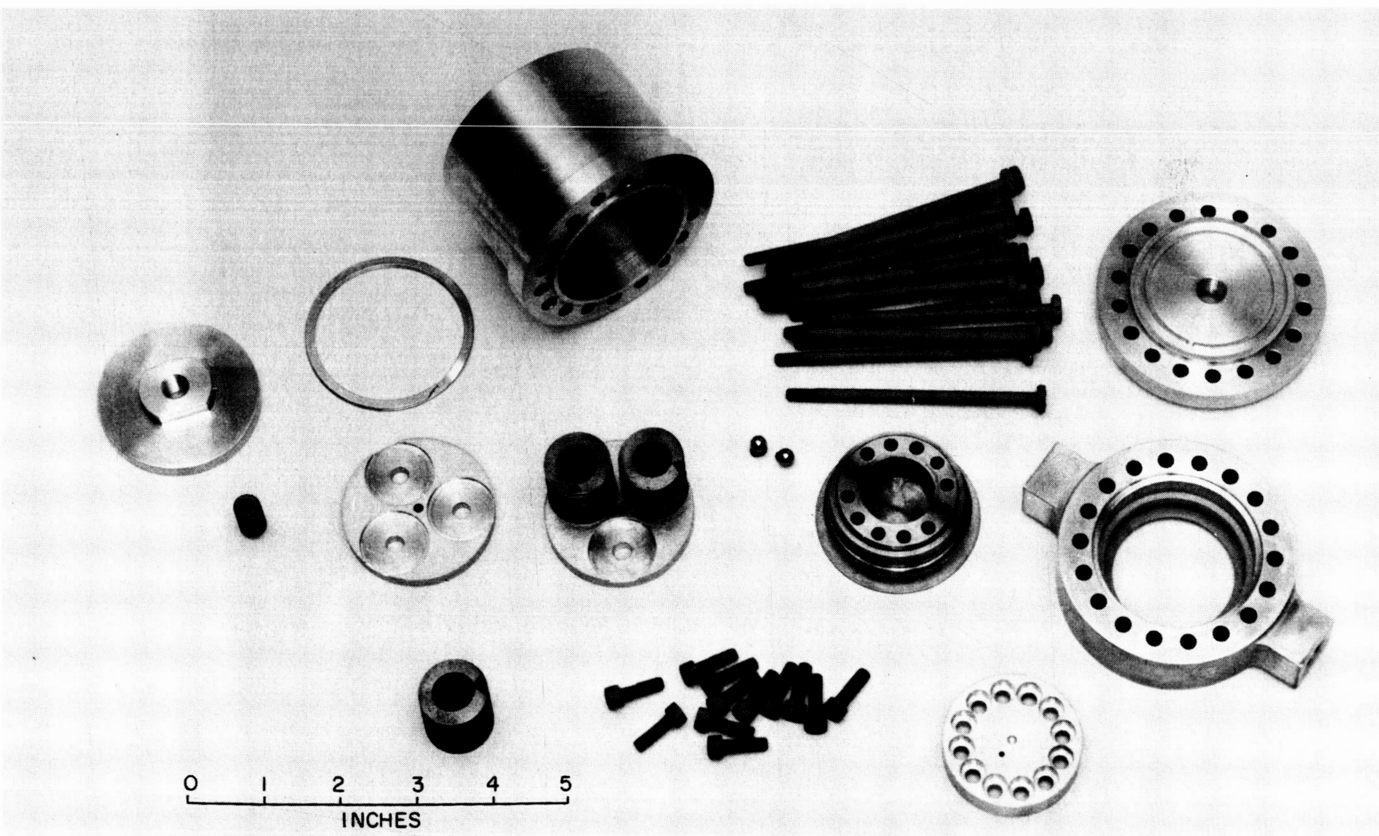

Fig. 11. Generant controller 3 parts

and rupture of a suspension ring. A suspension ring ruptured on the second unit before all the data could be obtained, which accounts for the blank spaces in Tables 6 and 7.

The suspension ring rupture was probably accelerated by seat vibration, which was intermittently present. Some evidence of erosion of the gold plating in the seat area was noticed.

VII. CONTROLLER 4

A. Description*

Figure 12 is an external view of controller 4. Figure 13 shows the parts and Fig. 14 is a cross-sectional drawing of controller 4 with the ball pushed on the seat. Figure 2 is a cross-sectional drawing of controller 4 with the ball pushed off the seat.

This is an all-metal (except seat ball), pressure-balanced controller with a ball seat. A $\frac{3}{8}$-in. ceramic ball is positioned on a $\frac{1}{4}$-in.-diam seat by an external yoke. A Belleville spring package bears against one face of the yoke to hold the seat ball in the open position. The opposite face of the yoke is the sensing port position which closes the seat ball when sensing pressure exceeds spring pressure. The sensing piston is sealed by a metal diaphragm which is supported by a backup ring.

*The general operation common to all four generant controllers is described in Section II. Table 1 gives drawing numbers, seat diameters and materials, etc., for each controller.

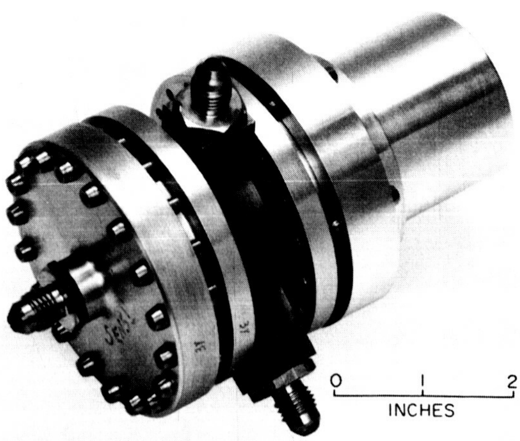

Fig. 12. Generant controller 4

The yoke consists of two stemmed pistons separated by three posts. The stems face toward each other and extend inward to support the backup rings and diaphragms which seal each end of the flow cavity. The ball seat is axially in line between the sealing diaphragms and is the same diameter as the effective diameter of the backup rings. Because the backup ring and seat areas are equal, variations in the upstream or downstream pressures do not tend to move the yoke.

The three backup rings not only support their respective diaphragms but also support and guide the axial travel of the yoke. The yoke operates between built-in stops. The backup rings function as a plurality of annular pivots thereby eliminating yoke journal friction.

The seat ball is positioned by a member piloted in the flow passage. The member is axially moved by the diaphragm and yoke.

The ball-positioning arrangements have been tested. Originally the ball was pushed off the seat by upstream flow and pushed on the seat by sensing pressure. The second arrangement, which requires installing the controller for flow in the reverse direction, was made so the ball would be pushed off the seat with a loose plunger and lowered onto the seat for closure.

The six 10-32 housing bolts are torqued to 25 in.-lb. The eighteen 6-32 sensing cap bolts are torqued to 15 in.-lb. Bolt heads and threads are lubricated with Apiezon L grease.

The housing and internal ball actuation parts are machined from 17-7 PH CRES and are heat-treated. The seat, which is integral with the housing, is diamond-lapped to conform to the seat ball. For hydrazine compatibility, the sensing port cap is made from 6061-T6 aluminum alloy. The yoke and other external parts are

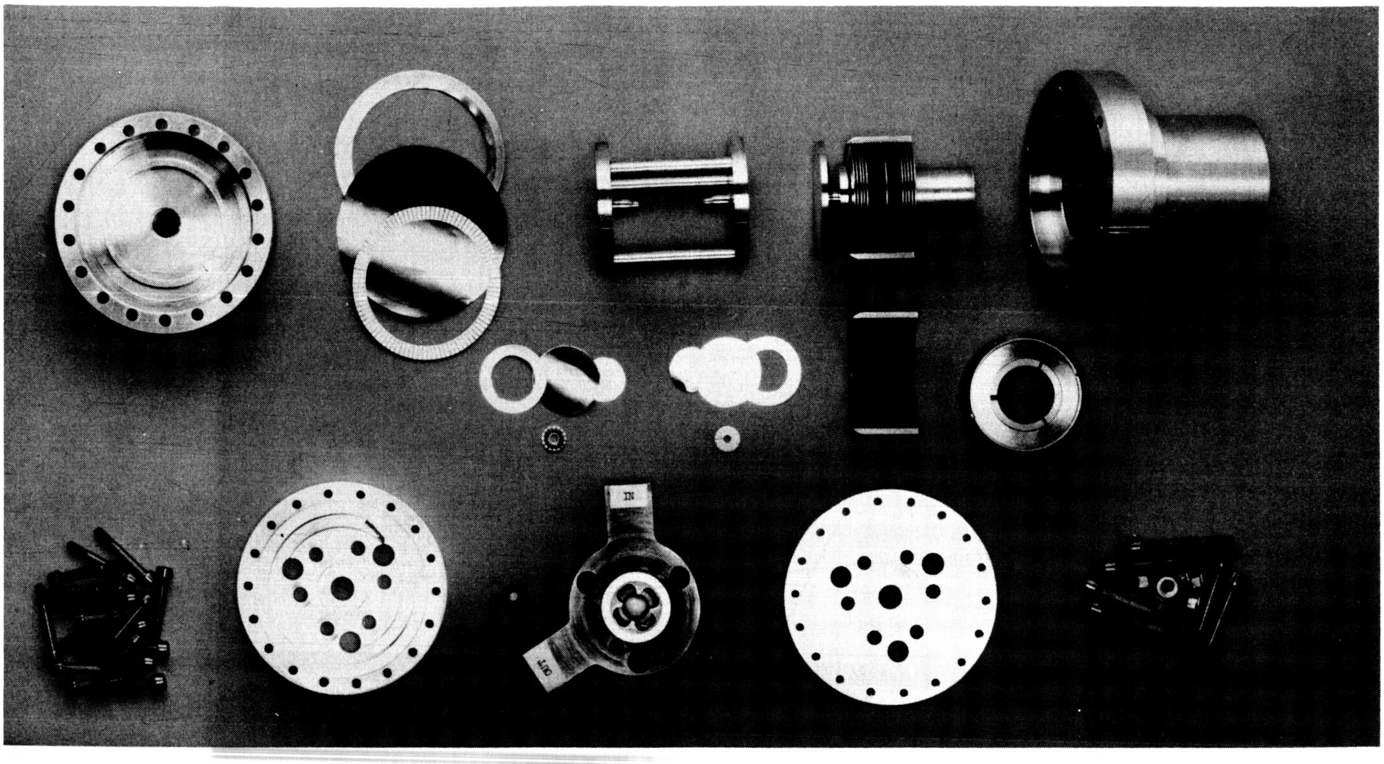

Fig. 13. Generant controller 4 parts

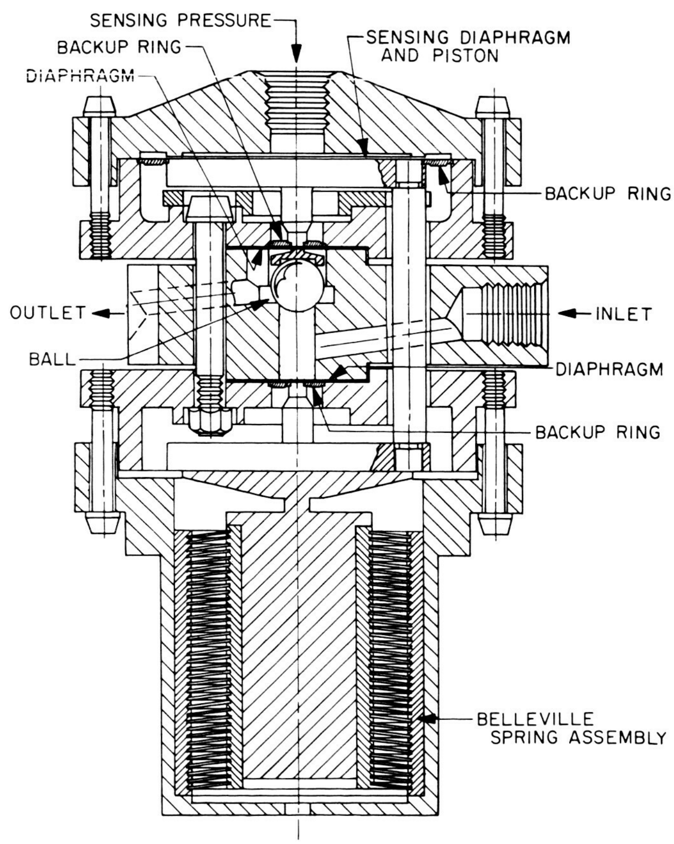

Fig. 14. Generant controller 4 cross section (ball pushed on seat)

machined from 7075-T6 aluminum alloy to provide strength, and particularly adequate compressive strength where the backup rings bear on the support annuli.

1. Type With Ball Pushed On Seat

In this version, shown in Fig. 14, the ball is located on the outlet side of the seat. To cause closure, the ball is pushed on its seat by a metal cap placed between the ball and the outlet diaphragm. Force for throttling or closure is applied to the ball through the diaphragm by the stem of the sensing pressure piston which is part of the actuated yoke.

A disadvantage to this arrangement is that excessive pressure applied to the sensing port would damage the controller by collapsing the external stem that seats the ball. Seat leakage during prolonged lockup could cause this destructive excess sensing port pressure.

2. Type With Ball Pushed Off Seat

In this version, the ball is located on the inlet side of the seat, as shown in Fig. 2. Upstream pressure tends to seat the ball. To assemble this model, the ball cap is removed and replaced with a 300 series CRES star spring to hold the ball on the seat. A hardened 17-7 PH CRES column with centering fins at both ends is placed in the seat outlet bore. The column extends from the bottom side of the ball to the outlet diaphragm. Spring-induced yoke movement unseats the ball by applying a force to the external stem which transmits that force through the diaphragm to the internal finned column which lifts the ball from its seat. Excess sensing port pressure buildup continues to retract the finned column after the ball has seated until the yoke engages its stop.

The advantage of this arrangement is that full upstream pressure can be applied to the sensing port without damage to the parts. The force required to push the ball off the seat, while it is under pressure, is the maximum load that can be applied to the ball. Thus, shock loading cannot harm the controller.

A possible disadvantage is the location of the finned column, which is directly in the flow stream. Since the ball is lifted off the seat only about 0.004 in., the restriction is at the seat and the additional ΔP caused by the finned column is negligible.

B. Performance

Both types (ball pushed on seat and ball pushed off seat) were tested in the setup shown schematically in Fig. 7. There was no difference in performance between the two types. Because of the freedom from damage in

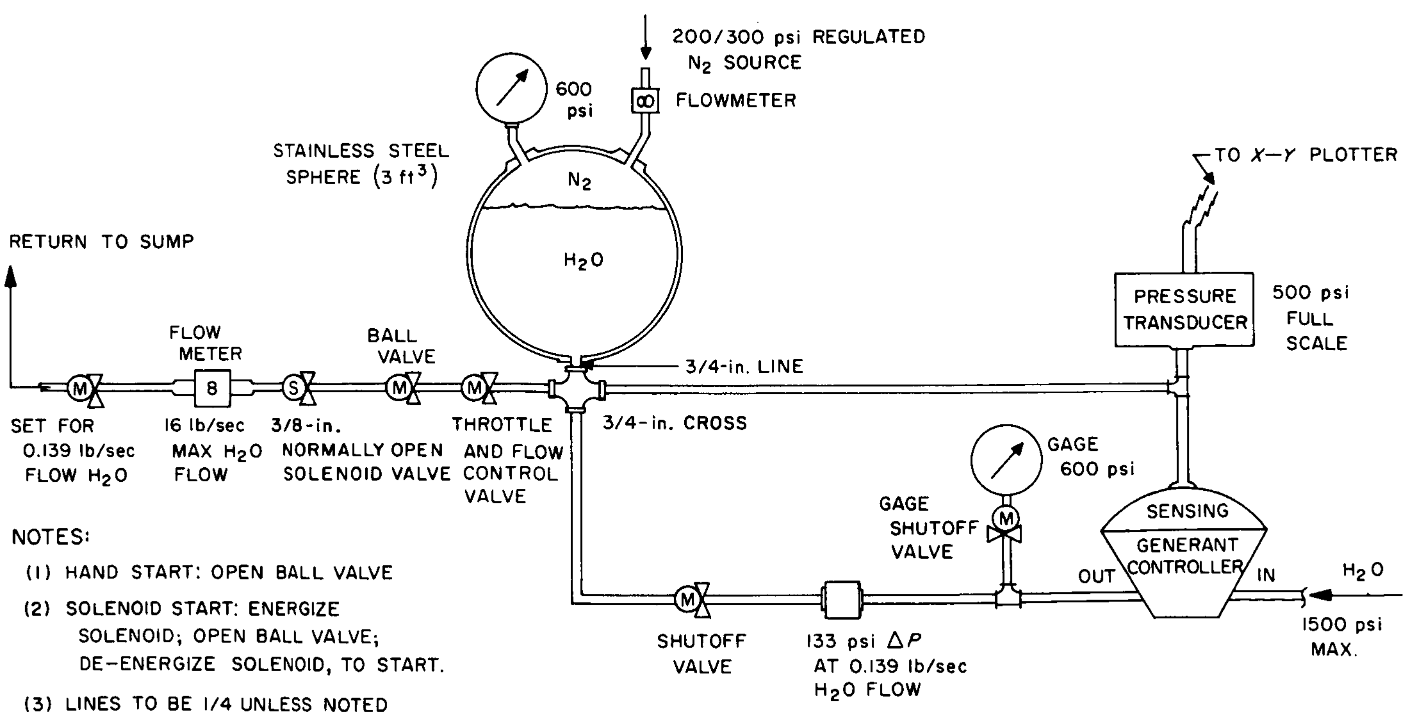

Fig. 15. Generant controller test setup for variable flow rate and simulated ullage runs

case of inadvertent overpressurization of the sensing port in the pushed-off type, the pushed-on type was reworked to the pushed-off type. Typical results of the above tests are recorded in Tables 2–7.

This latter type of controller was also tested in the setup shown schematically in Fig. 15. Results of these tests, which simulate tank expulsion under various conditions, are shown on the *X–Y* plots of Figs. 16–19.

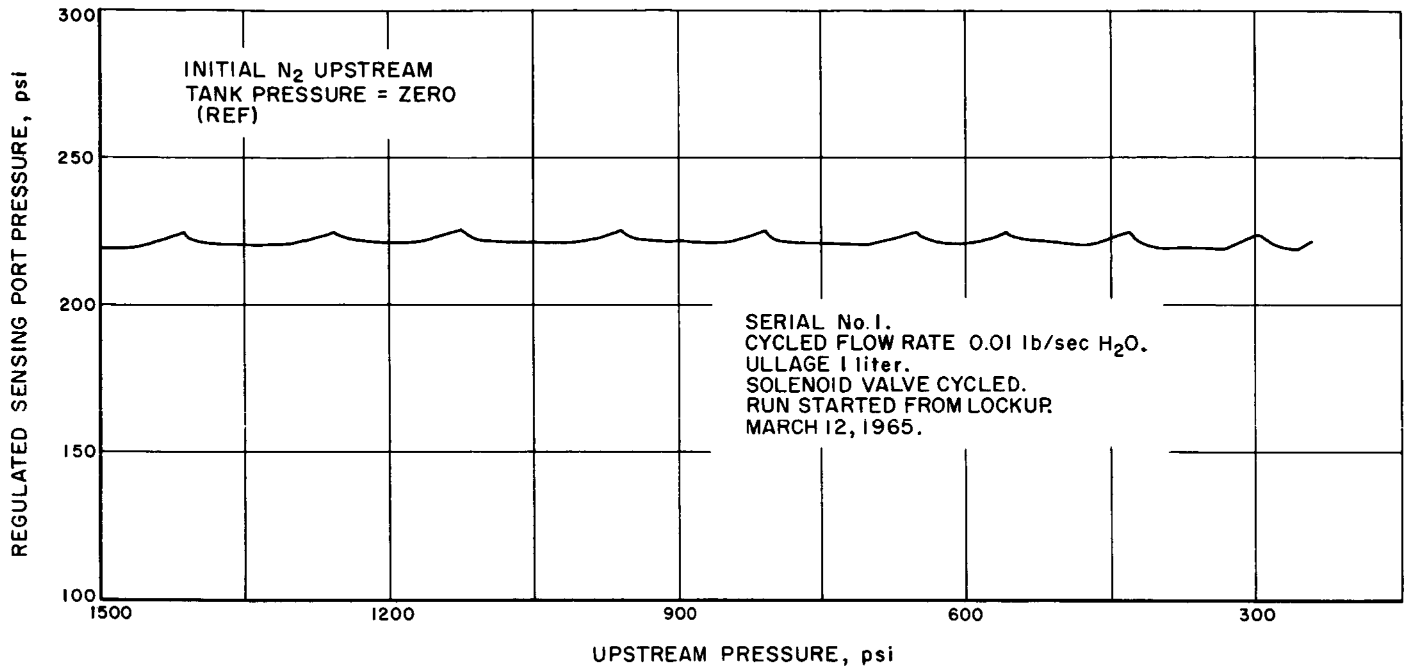

Fig. 16. Generant controller 4 blowdown run simulating cycled downstream tank expulsion. Low ullage and low flow

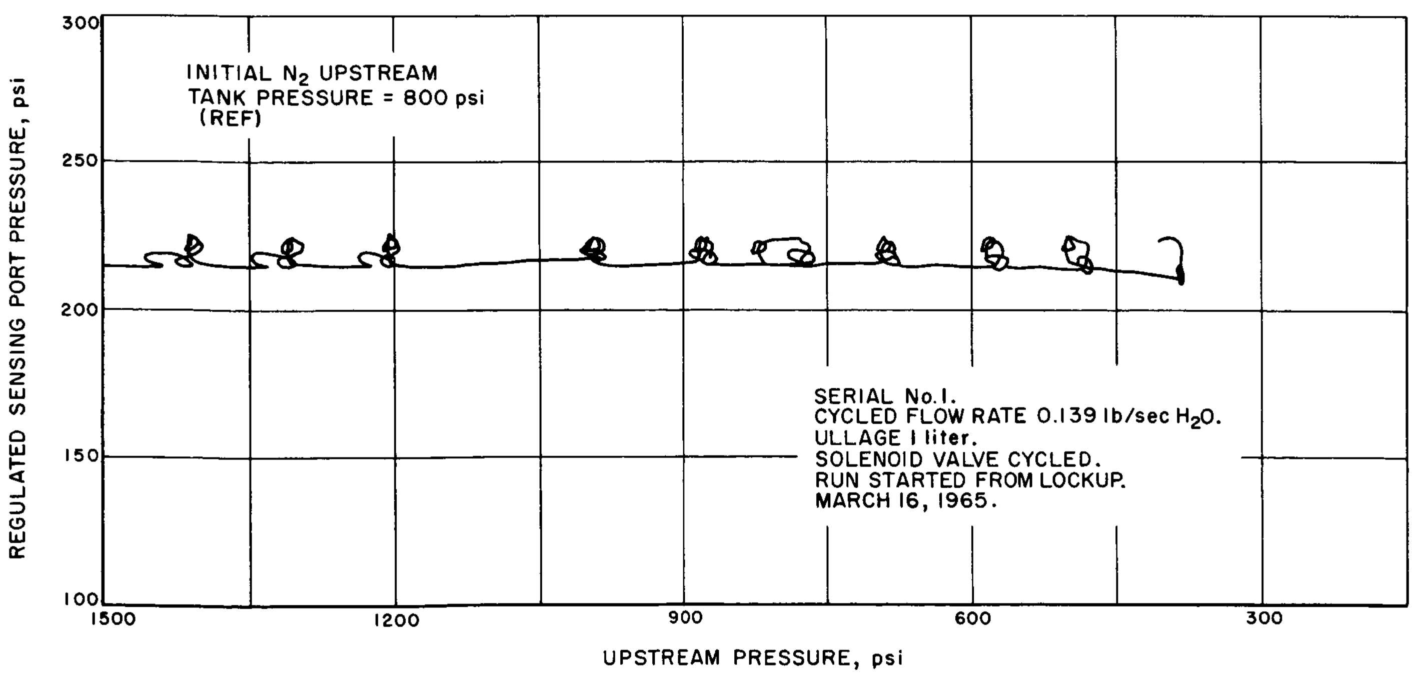

Fig. 17. Generant controller 4 blowdown run simulating cycled downstream tank expulsion. Low ullage and high flow

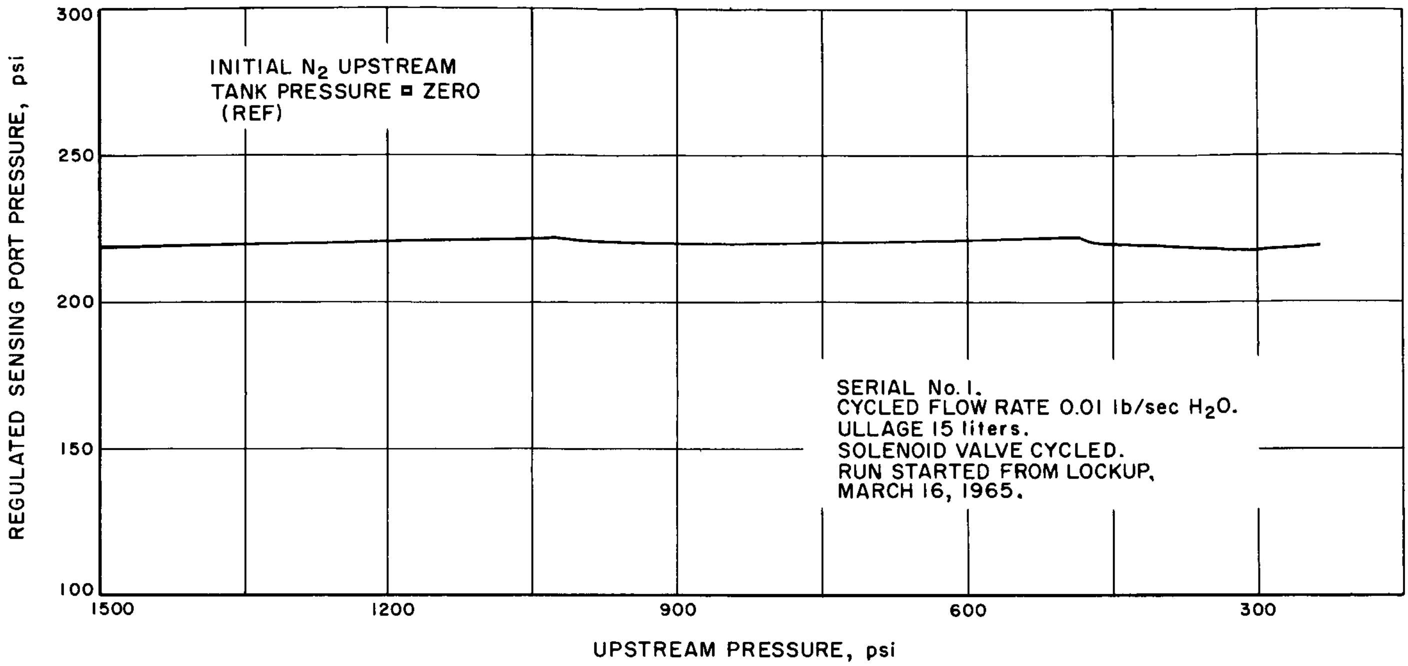

Fig. 18. Generant controller 4 blowdown run simulating cycled downstream tank expulsion. High ullage and low flow

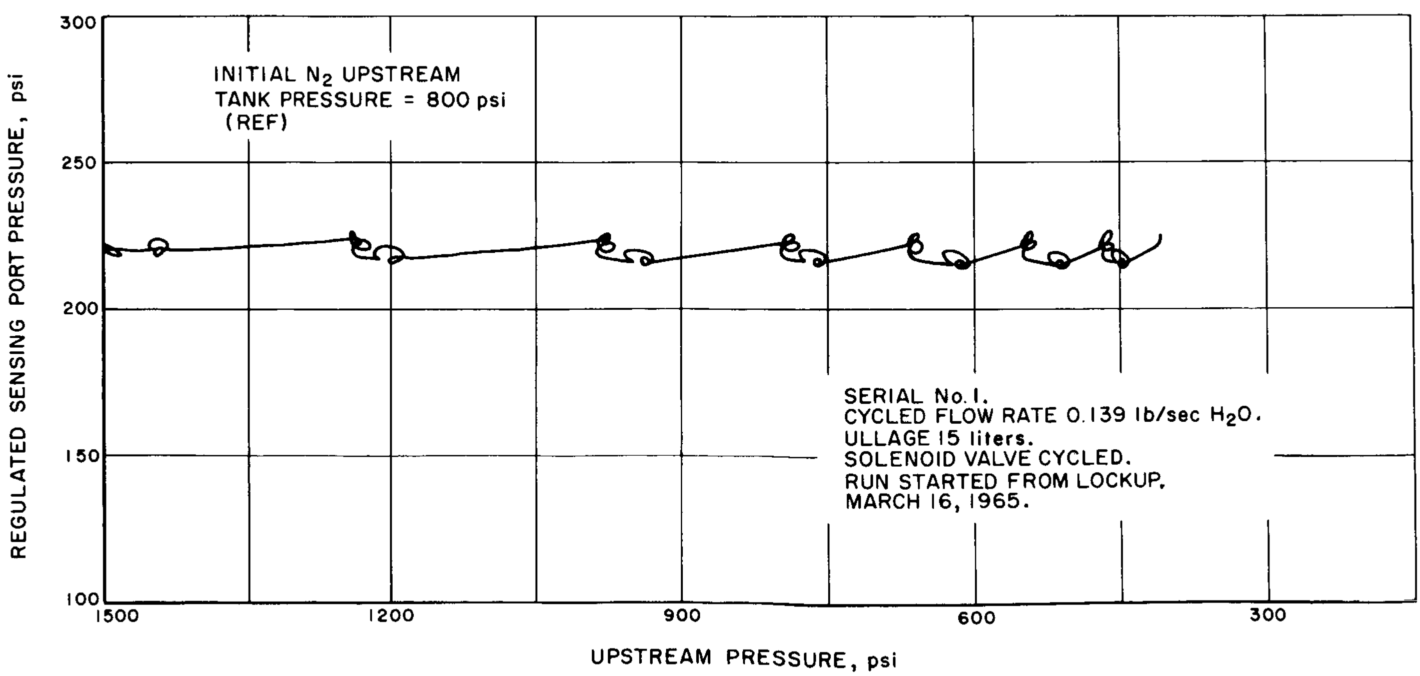

Fig. 19. Generant controller 4 blowdown run simulating cycled downstream tank expulsion. High ullage and high flow

Figure 20 is the X–Y plot of a typical blowdown run in the test setup of Fig. 7. Pressure regulation is within 0.75 psi, from 1350-psi upstream pressure to tailoff.

Explanation of the Tables and Figures and comments on the controller characteristics are included in Section VIII.

In addition, two controllers were prepared for use in the *Mariner* '66 Feasibility Study (see Appendix). Because the *Mariner* flow requirement was 59% greater than the maximum ALPS flow rate, the *Mariner* seat was enlarged to 0.344-in. diameter to receive a 0.500-in. ball (ALPS had a 0.250-in. seat and a 0.375-in. ball). Regulation, during hot firing blowdown tests, was within 1 psi.

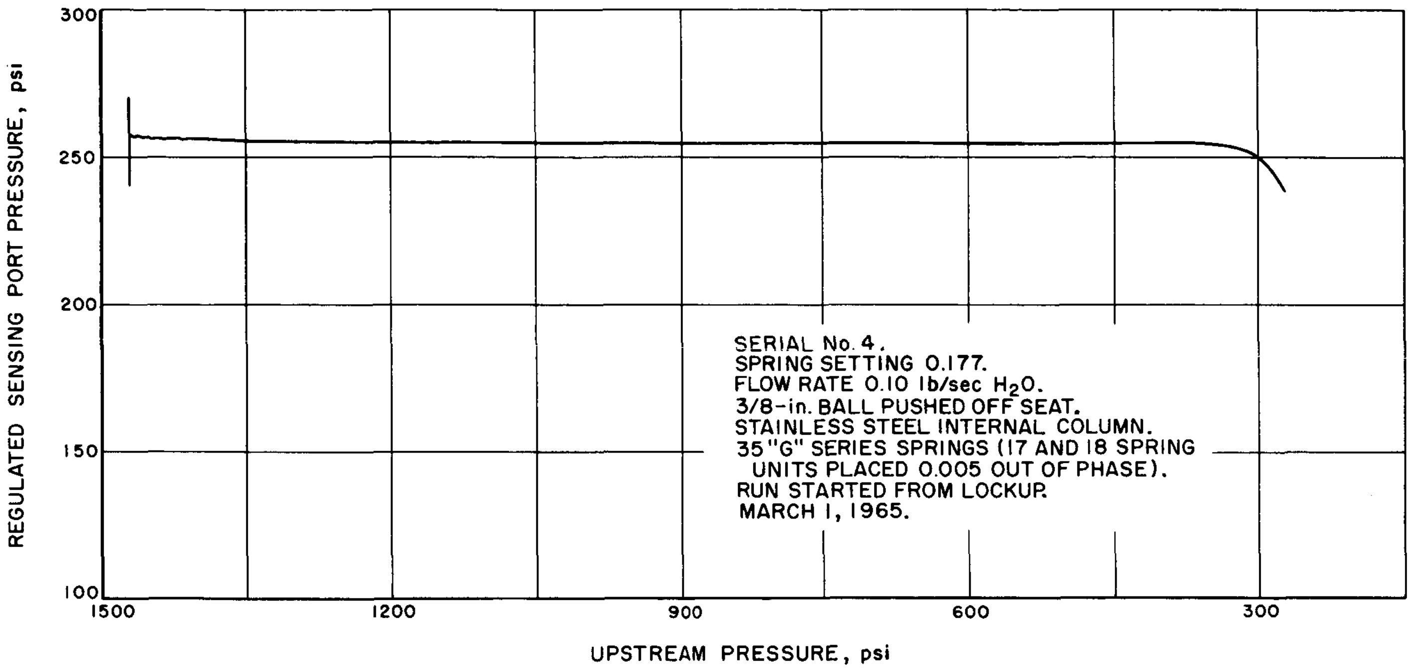

Fig. 20. Typical generant controller 4 blowdown run as recorded on X–Y plotter

VIII. TEST RESULT SUMMARY OF GENERANT CONTROLLERS

All tests, with exception of the proof test, the external leak test, and some of the flow tests, were conducted with nominal 5-μ-filtered distilled water. Water contamination problems, which were in evidence during the early phase of the testing, were solved as the tests progressed.

Controller 4 is superior to the others from a standpoint of control accuracy, durability, and ease of servicing. Among the knife edge seat group, controller 1 was better than controller 2 or 3. Controller 3 failed before preliminary tests could be completed but the ruptured support ring, which caused the failure, was the forerunner of the highly successful backup ring used in controller 4.

No attempt was made to secure leak-tight lockup of controllers 1, 2, and 3 because of early contamination damage to the seats and because even if good lockup were obtained, operating accuracy was not satisfactory. Controller 1 is capable of good lockup but satisfactory lockup of controllers 2 and 3 would be very difficult to obtain because of the large-diameter seats. The knife edge seats were susceptible to contamination damage.

Using water as the test medium, the ball seat propellant leakage of controller 4 was less than the allowable 0.1 cm^3/hr at 1500 psi. Tests have been recorded during which water leakage became undetectable after the initial fifteen minutes of slight seepage. This 17-7 PH seat has endured many test runs without any appreciable degradation.

During testing in the setup of Fig. 7, all four controllers exhibited a high frequency vibration or squeal at various points in the flow range. With the knife edge regulators (1, 2, 3), the tendency to vibrate was least in 1 and greatest in 3, thus indicating vibration tendency in proportion to seat diameter.

To combat this vibration, a No. 80 (0.0135-in.-diam) orifice was installed at the entrance to the sensing port of controller 3. The orifice was too large to be effective. A needle bleed valve was placed in series with the orifice and flow was throttled until the vibration ceased. This hydraulic damping was effective but the equivalent orifice was exceedingly small. The needle valve was removed from the line and calibrated. For reference, the water flow rate through the valve at the setting that eliminated squeal was 0.0011 lb/sec at 250-psi ΔP.

Controller 4 tended to vibrate at low flow when upstream pressure exceeded 1200 psi. A No. 78 (0.016-in.-diam) orifice placed in the sensing port reduced but did not eliminate the seat squeal. When controller 4 was tested in the setup of Fig. 15 (simulated ullage runs), the vibration completely disappeared and did not reappear when the No. 78 orifice was removed. This sequel is associated with the peculiarities of the test setup.

No problems were experienced during the nitrogen-proof and external leak testing of the controllers.

The flow rate versus sensing pressure data of Tables 2 and 3 and the outlet ΔP versus sensing pressure data of

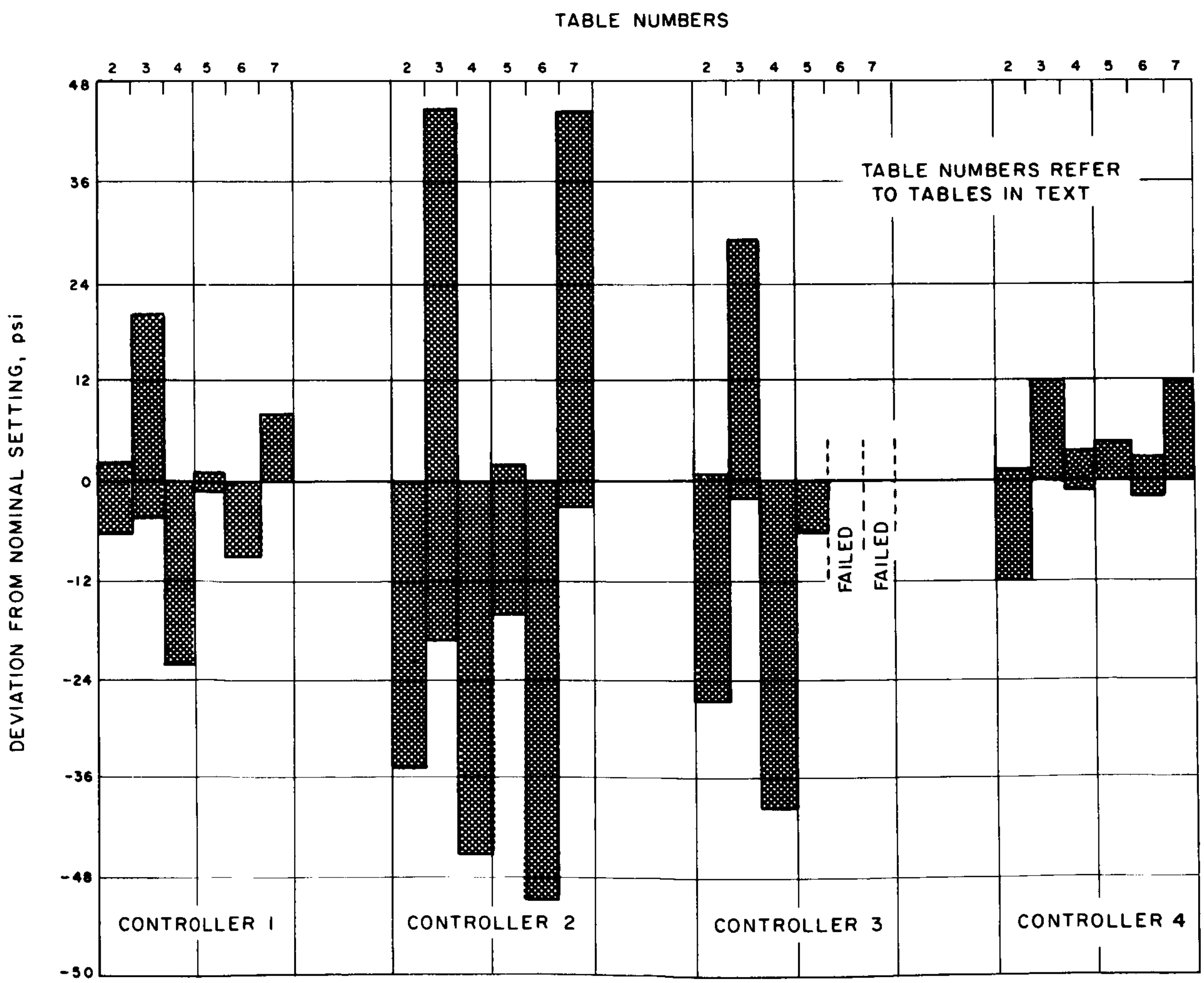

Fig. 21. Composite of Tables 2–7 showing comparative generant controller deviation from nominal setting

Tables 4–7 were obtained from tests conducted with all four controllers in the test setup of Fig. 7. All the above tests were conducted with valves F and G (Fig. 7) open, and valve E was used to control the stepped flow rate. Data from Tables 2 and 3 were obtained with valves D wide open. The ΔP of Tables 4–7 was obtained by throttling valves D to give the desired increased pressure on gage B. This ΔP simulates the variation in system pressure drop that is anticipated during a hot run and also indicates how well the controller is pressure-balanced. All sensing port pressures were obtained by visual reading of gage C. The deviation from nominal sensing pressure of each controller is listed at the bottom of each table and gives an indication of the relative controller sensitivity.

Figure 21 is a visual composite of Tables 2–7 showing comparative deviation of each of the four controllers from the nominal sensing pressure setting.

Only controller 4 was tested in the Fig. 15 test setup for simulated ullage runs. The purpose of the Fig. 15 test setup is to provide an ullage system scaled so that the time required for the controller, at full flow of water, to produce a given pressure change within the ullage will be identical to the time required for the controller, during a hot run at full liquid flow, to generate the gas to produce the same pressure change in the ullage of the full-scale ALPS propellant tank.

Using the setup shown in Fig. 15, nitrogen ullage was pressurized to the controller regulated pressure and then the shutoff valve was opened and the controller was allowed to lockup. The run could be started either manually (ball valve) or electrically (solenoid valve). The valve downstream of the flow meter serves as a flow limiter and the flow rate is set by the throttle and flow control valve. The valve at the controller "out" port is set to simulate the injector ΔP. The nitrogen ullage volume can be set anywhere between 1 and 15 liters to simulate the full-scale tank range from full to empty.

Figures 16–19 are X–Y plotter records from tests which simulate variable expulsion under conditions of full tank (low ullage) with high and low flow and nearly empty tank (high ullage) with high and low flow. Variable expulsion was simulated by a solenoid valve cycling the flow from the tank with the same frequency as is required for tank pressure equilibrium to be reached after each flow actuation. The upstream tank pressurization notation on Figs. 16–19 refers to the controller water supply system (not shown) which is tailored to equalize the time required for the blowdown run at high and low flow. The circles on the high-flow plots (Figs. 17 and 19) were recorded when flow was started or stopped; they are caused by a combination of water hammer and pressures equalizing in the circuit. These tests (Figs. 16–19) were conducted with the upstream filter removed. In earlier tests with the filter installed, the added pressure drop caused the actuation circles to elongate into rectangles on the upstream pressure axis. However, the sensing pressure axis was unchanged.

When flow is started from lockup, the controller remains closed and the flow is supplied by water in the tank which is expelled by the ullage pressure. When ullage pressure drops, the controller opens and replaces the water in the tank while maintaining the flow through the metering section until lockup. The cycle of first the tank and then the controller furnishing the flow continues throughout the run. During a variable flow blowdown run, sensing pressure varies about 5 psi at low flow and about 10 psi at high flow.

The solenoid-actuated runs of Figs. 16–19 were also conducted with hand valve actuation. The X–Y plots for solenoid and for hand valve actuation were identical.

IX. DEVELOPMENT OF MACHINE ELEMENTS LEADING TO THE CONTROLLER 4 CONFIGURATION

Several of the machine elements utilized in generant controller 4 were developed separately. These ALPS developments are of interest for components other than controllers. For example, a Belleville spring package and a backup ring were used in the *Ranger* and *Mariner* midcourse propulsion-system gas-pressure regulators. Four of these machine elements are discussed in this Section.

A. Belleville Spring Package

1. Belleville Spring Characteristics

A Belleville spring is a ring stamped from sheet material and is dished to the shape of a shallow cone. It is used in applications where deflections are small. By stacking them in parallel, high load capacity can be attained.

The spring is deflected by pressing it through the flat position to a conical position in the opposite direction while being supported at the inner and outer peripheries. The load-vs-deflection curve is dependent upon the cone angle. When H/T (axial height of dish-vs-material thickness) is 1.5, there is a flat spot in the upward sweep of the load-vs-deflection curve such that the load remains constant while deflection continues. When H/T is greater than 1.5, the upward sweep of the load-vs-deflection curve rounds over and proceeds downward before resuming the upward trend. It is operation along this negative or back slope (Fig. 22) of the curve that produces the best range for giving the flattest regulation to generant controllers.

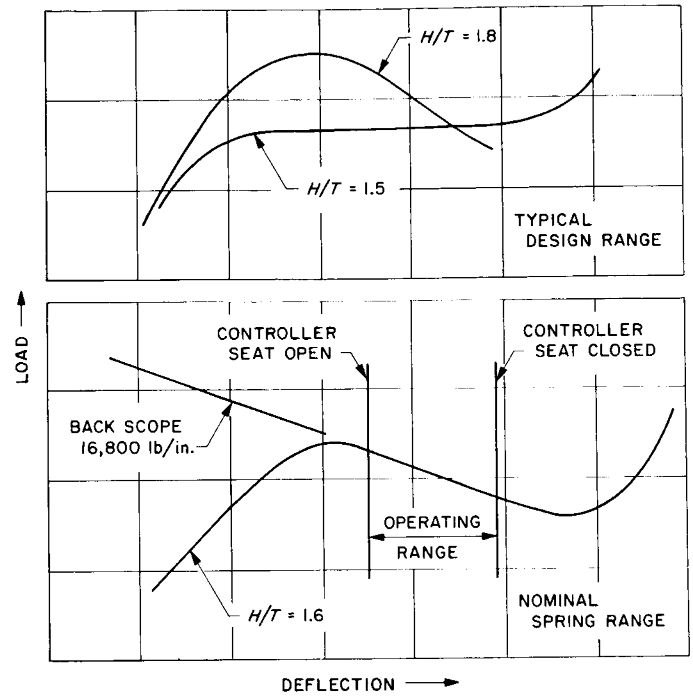

Fig. 22. Typical and nominal Belleville spring for generant controller

2. Friction or Hysteresis Reduction

The original spring package consisted of 20 Belleville springs stacked in parallel and centered on the OD by a sleeve and on the ID by a shaft. In order to reduce the friction, the springs were separated by rings rolled from round music wire. It was found that hysteresis was excessive because of high bearing loads, which resulted from the additive loads of each spring being carried through the wire rings.

To assure that each spring carries only its own load, a tube was bored with shallow grooves to support and space the OD of the spring. In a similar manner a smaller tube was grooved on the OD to support and space the ID of the spring. For assembly, the sleeves were split axially into several segments to permit insertion of the springs. The segments were maintained in position on the springs by inserting a shaft through the inside segments and by placing the entire package of 20 springs into a bored housing.

This spring package (Fig. 13) reduced the hysteresis to an acceptable level. Two or more spring packages can be placed in parallel (end to end) to provide higher loads.

3. Controlling Characteristics of Load-vs-Deflection Curve

The Belleville springs used in the generant controller are made from a standard AISI 6150 alloy steel spring which is nominally 1.56-in. OD × 1.00-in. ID and is stamped from 0.020-in.-thick steel sheet and dished 0.030 in. (0.050-in. total height).

The spring cone angle can be changed by clamping the spring on a mandrel to the desired angle and reheat-treating, quenching and drawing while in the clamped position. Reducing the angle lowers the spring force and decreases or eliminates the back slope. Increasing the angle raises the spring force and steepens the backslope of the load-vs-deflection curve.

The bore and OD of all springs are sized by grinding to provide closer tolerances and concentricities. The springs are then barrel-tumbled to break the sharp edges and are silver-plated to assist dry lubrication and to resist corrosion. The grooves of the segments that support the springs are dry-lubed with a 0.0003-in. coating which consists of powdered molybdenum disulfide in a phenolic binder.

Even though the springs from a batch appear to be identical, the load-vs-deflection curve for each spring is different (Figs. 23 and 24). A spring package prepared from a matched set of individual springs will have the identical load-vs-deflection curve as the individual spring if the load scale on the plotter is changed to give the same height. The steepness of the back slope can be reduced by random mixing of the springs comprising the package or by using an inside or outside spacer between two matched sets in parallel so that one set picks up the load before the other (Fig. 25).

4. Trend of Future Development

By actual measurement, the OD increases 0.0033 in. and the ID decreases 0.0033 in. when the spring is deflected through its flat position. This radial growth produces sliding friction between the springs and the holder and accounts for much of the hysteresis. If the spring is supported on long flexible columns (every 90 deg on both OD and ID), the friction is removed elastically. This method of support is used in obtaining the load-vs-deflection curve for each individual spring.

Future developments will be in the direction of applying the four-point suspension principle to the spring package for the purpose of further reducing hysteresis.

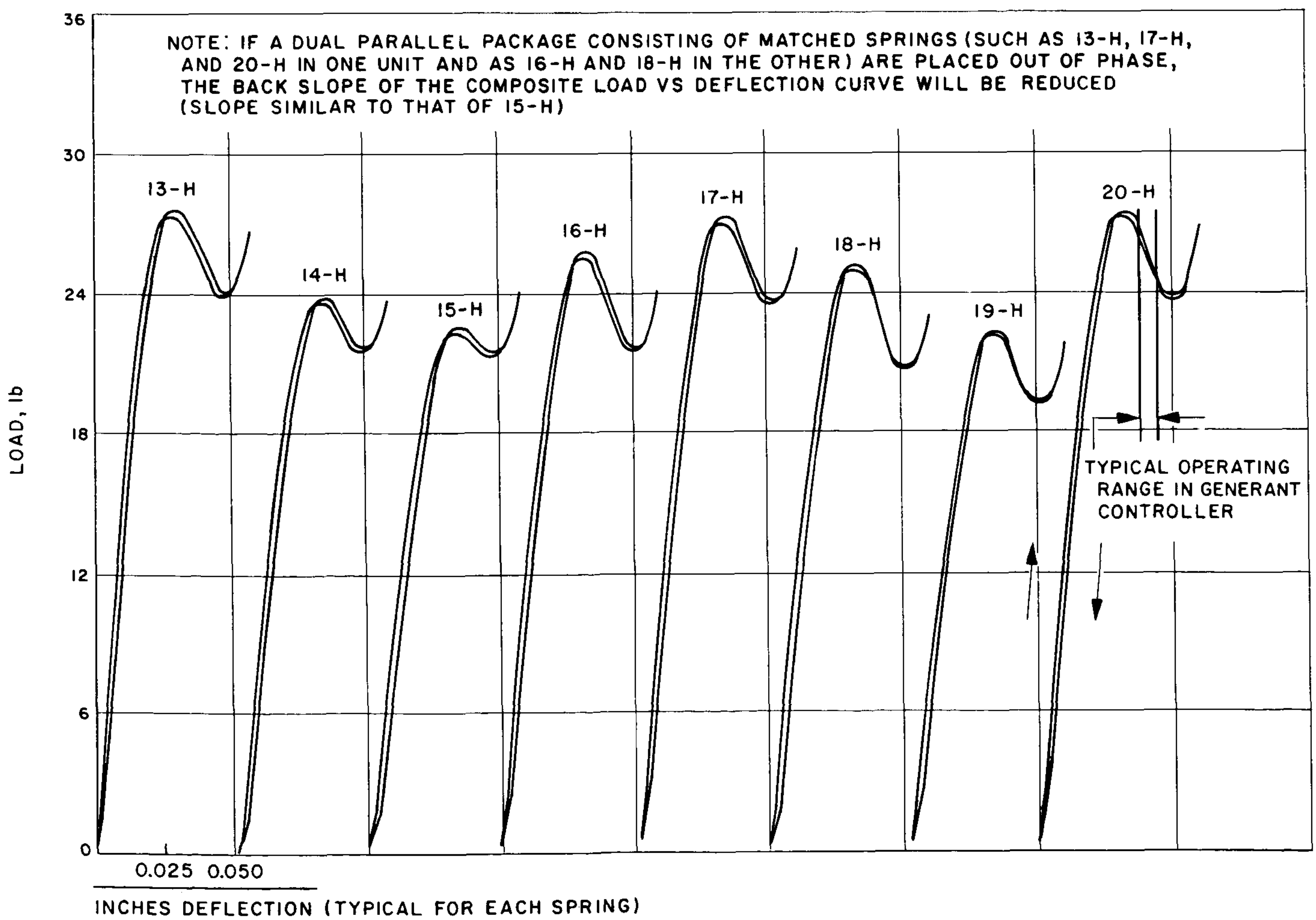

Fig. 23. Typical variation of load vs deflection for Belleville springs

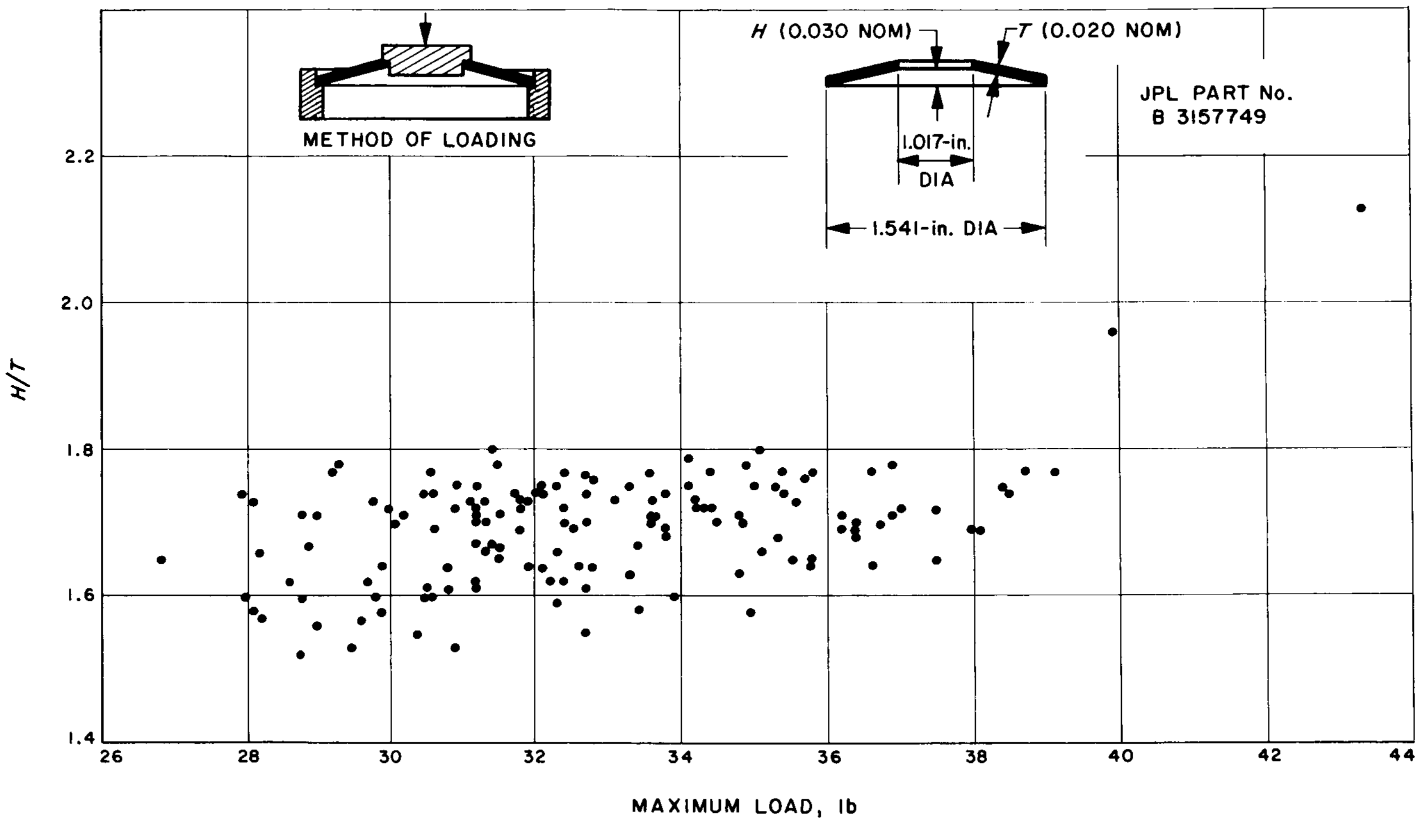

Fig. 24. Typical scatter for a batch of 150 Belleville springs

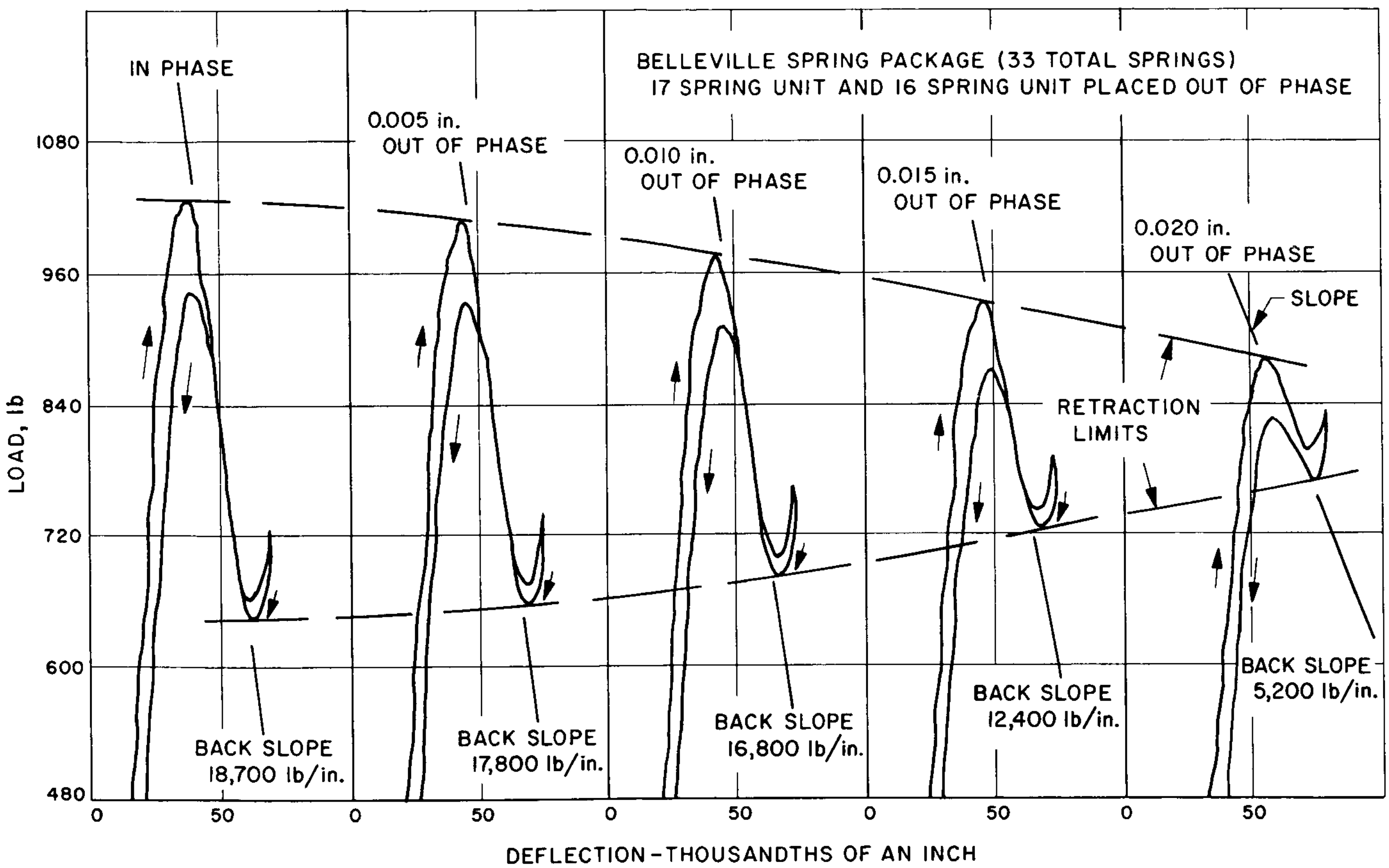

Fig. 25. Effect of slope and retraction limits when Belleville springs are out of phase

B. Backup Ring

1. Definition and Description

The backup ring (Fig. 13) serves the dual function of radially supporting a central reciprocating member and supplying the structural backing for a diaphragm placed across the face of the reciprocating member.

The backup ring is placed in the annular gap between the stationary and the reciprocating members. The ring face is flush with the faces of the adjacent members and is supported on narrow lands produced by machining a step in each mating member. The ring is radially slit to form a plurality of pie-shaped segments. The segments are joined together at the ID by a very shallow band formed by controlling the slitting depth. The diaphragm is statically sealed by being clamped at its outer periphery to the stationary member. The slitting enables the ring to assume a conical shape with the application of very little force. Thus each segment of the backup is able to flex with the reciprocating member.

2. Design

The backup ring cross section is contoured with the intent of obtaining uniform stress distribution. The annuli of support (inner and outer peripheries) of the ring are thinned to carry the load in shear. The center section is thickened to sustain the bending load. The rings have been machined from 7075-T6 aluminum alloy.

The effective area is critical when a backup ring is used to create a force balance. This area is defined as the area which, if uniformly loaded by a unit pressure, would exert the identical total force as the moving backup ring and piston configuration would exert when subjected to the same unit pressure (Fig. 26).

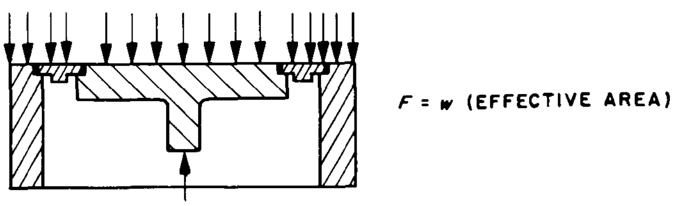

Fig. 26. Definition of backup ring effective area

The center-of-gravity computation for the backup ring sector (Fig. 27) is

$$b = 38.197 \frac{(R^3 - r^3)}{(R^2 - r^2)} \frac{\sin \alpha}{\alpha}$$

$$\lim_{\alpha \to 0} \frac{\sin \alpha}{\alpha} = \frac{\pi}{180} = 0.0174533$$

$$b = 38.197 \left(\frac{(R^3 - r^3)}{(R^2 - r^2)} \right) 0.0174533$$

$$b = 0.6666667 \left(\frac{(R^3 - r^3)}{(R^2 - r^2)} \right) = \frac{2}{3} \left(\frac{R^3 - r^3}{R^2 - r^2} \right)$$

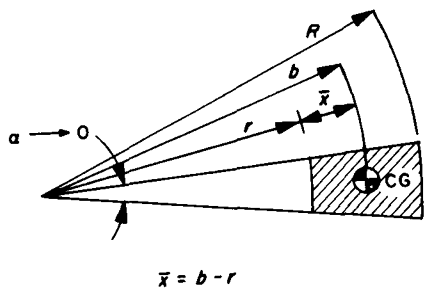

Fig. 27. Center of gravity computation for backup ring sector

The backup ring summation of moments (Fig. 28) is

$$R_R - R_2 = R_1$$

$$\bar{x} R_R = R_2 (R - r)$$

$$\frac{\bar{x}}{R - r} = \frac{R_2}{R_R}$$

By using the center of gravity and summation of moments, the effective backup-ring diameter can be shown to be

$$D_E = 2 \sqrt{R^2 - \bar{x}(R + r)} \qquad (1)$$

from which the effective area is

$$A_E = \pi [R^2 - \bar{x}(R + r)] \qquad (2)$$

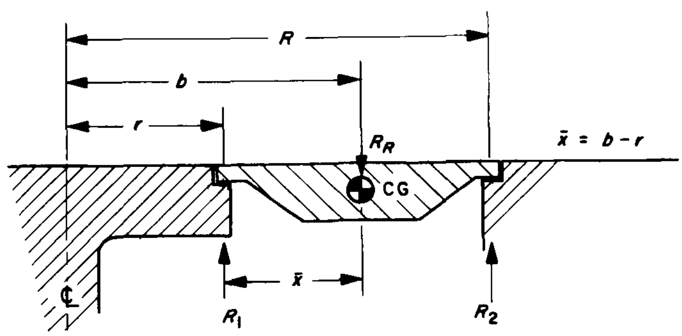

Fig. 28. Backup ring summation of moments

where

D_ε = effective diameter
A_ε = effective area
R = outside radius
r = inside radius
$\bar{x}$ = radial distance from ID of backup ring to sector CG

The following forms of Eq. (1) are more convenient for sizing the backup ring:

$$R = r\left[\frac{\sqrt{3}}{2}\sqrt{\left(\frac{D_\varepsilon}{r}\right)^2 - 1} - \tfrac{1}{2}\right] \tag{3}$$

and

$$r = R\left[\frac{\sqrt{3}}{2}\sqrt{\left(\frac{D_\varepsilon}{R}\right)^2 - 1} - \tfrac{1}{2}\right] \tag{4}$$

For the special case when $r = 0$, the values of $\bar{x}$ and R are given below.

From Fig. 27,

$$\bar{x} = b - r$$
$$= \frac{2}{3}\left(\frac{R^3 - r^3}{R^2 - r^2}\right) - r$$

When $r = 0$,

$$\bar{x} = \frac{2}{3}R \tag{5}$$

From Eq. (3),

$$R = r\left[\frac{\sqrt{3}}{2}\sqrt{\left(\frac{D_\varepsilon}{r}\right)^2 - 1} - 0.5\right]$$

By placing r inside the brackets,

$$R = \left[\frac{\sqrt{3}}{2}\sqrt{r^2\left(\frac{D_\varepsilon}{r}\right)^2 - r^2} - 0.5\,r\right]$$
$$= \frac{\sqrt{3}}{2}\sqrt{D_\varepsilon^2 - r^2} - 0.5\,r$$

When $r = 0$,

$$R = \frac{\sqrt{3}}{2}D_\varepsilon \tag{6}$$

3. Multiple-Piece Unit

The one-piece ring previously described is sufficiently flexible if the peripherial diameter is relatively large.

Small-diameter rings are excessively stiff. To overcome this objection, each segment is scribed with an identifying letter and the joining web of each segment is broken. To facilitate final assembly in the controller, the segments are bonded together in the following manner. The segments are assembled in order in a jig. Self-vulcanizing rubber paste is applied to the back surface of the segments, so that after curing, the rubber-bonded ring can be removed from the jig in one piece. The rubber-bonded backup ring results in a unit of maximum flexibility.

4. Journal Friction Elimination

For small amounts of travel, a backup ring provides a very low spring rate support that is nearly friction-free. Each backup-ring segment acts as an individual toggle in providing radial support that is immune to seizure. While not absolutely necessary, lubrication is applied to the support annuli to reduce wear and friction to a minimum.

5. Pressure

A 7075-T6 aluminum alloy backup ring, which was seated in 6061-T6 aluminum alloy support annuli and which was covered by a 0.003-in.-thick 1100-0 aluminum diaphragm, has been proofed to 5400 psi without damage.

6. Trend of Future Development

Reduction of hysteresis will be the objective of future development. There are three ways, which could be used separately or together, presently apparent for reducing backup-ring friction.

One method is to make the ring from a high-strength material so that the thickness at the support annuli could be reduced. During flexure this would reduce the amplitude of frictional contact with the covering diaphragm at the support annuli.

Another method is to provide an elastic outer support annulus (such as a rectangular column for each ring segment) to eliminate the radial sliding of the segments in the support annulus during flexure.

The third method is to harden the support annuli bearing areas and the backup ring so that the bearing area can be reduced. This reduction in area is reflected in a narrower support annulus which results in a reduced effective diameter shift during flexure. Hardened surfaces in contact also tend to reduce friction and wear.

C. *Metal Diaphragms*

1. Aluminum

The original diaphragms were made of 0.003-in.-thick 1100-0 aluminum sheet. The reason for using soft aluminum was threefold. Firstly, soft aluminum eliminated the need for a separate gasket because it is soft enough to form a good seal in compression. Secondly, pure aluminum was compatible with propellants and with water. Thirdly, soft aluminum could best resist flexure beyond the yield point which occurs in the diaphragm where it overlays the support annuli of the backup ring.

The use of aluminum for diaphragms was abandoned because of the development of pinhole leaks during water tests. The cause was believed to be galvanic action originating with foreign particles embedded in the diaphragm surface. The edges of the pinholes were always black.

2. Stainless Steel

The present diaphragms (Fig. 13) are made from 0.001-in.-thick Type 302 stainless steel shim stock. Because of the greater strength of this hard-rolled material, the thickness was reduced over that previously used. The thin material is adequate to span the unsupported slots in the backup ring and is better able to withstand flexure. Soft aluminum gaskets, cut from 0.003-in.-thick aluminum sheet, are used to seal the diaphragm periphery. Since steel is further down the electromotive list than aluminum, no diaphragm corrosion problem exists.

3. Backup Disc

Diaphragm life has been greatly extended by placing a 0.001-in.-thick stainless steel shim stock backup disc between the backup ring and the diaphragm. The backup disc is free-floating and is slightly larger in diameter than the OD of the backup ring.

4. Lubrication

A light film of Apiezon L grease is applied to both sides of the gaskets, backup discs, and diaphragms.

Gaskets can be re-used but should be lubricated before each use.

D. *Ball and Seat*

1. Ball

Sapphire or ceramic balls were selected in place of steel balls because of their light weight, corrosion resistance, compatability witl propellants, extreme surface hardness, and availability with high spherical accuracy and fine surface finish. For this application, balls having a spherical accuracy of 0.000025 in. and a surface finish of 1 μin. RMS were selected.

A ball is prepared for service by rubbing the ball on a rubber mat with a block of Micarta. The Micarta is dimpled to retain the ball and silver polish is applied to the mat. This scrubbing action cleans the ball and removes any metal pickup that might be gained on the surface.

A light film of Teflon is applied to the ball by using a dimpled block of Teflon to rub the ball in a circular motion on a clean piece of paper.

The ball is half-submerged in alcohol and sable-brushed to remove lint. The ball is transferred while still wet, with tweezers, to its final resting place on the seat.

2. Diamond Laps

The laps for preparing the seats are made by rolling a steel ball (same size as seat ball) between two sintered carbide plates on which the diamond compound has been spread. After diamond impregnating, the ball is soft-soldered to a brass handle. The diamond compound for the roughing lap is 15μ and 1μ for the finishing lap.

3. Seat Preparation

The seat is rough-lapped, using the laps prepared as above, to form a 0.003-in. land using way oil as a lubricant. The land is smoothed using the finishing lap and way-lube or kerosene as a lubricant.

After cleaning the seat, a light film of Apiezon L grease is transferred to the seat by contact with a lightly lubed ball cemented to a handle. The seat is then ready to receive the seat ball.

X. CONCLUSIONS

For a straight blowdown run at a constant flow rate, friction in the controller is not critical because ball travel is in the same direction throughout the run and friction is essentially constant. For this class of service, the present state of development of controller 4 is adequate.

For the ALPS application, which requires a variable flow rate, controller friction is critical because the seat ball actuation yoke reciprocates during regulation and each change in direction is resisted by the controller friction. The friction present in controller 4 causes a 5- to 10-psi regulation variation during a variable flow rate blowdown run.

The Belleville spring package and the backup ring are two areas in the controller where friction could be reduced. Suggestions for reducing friction in these areas are included in Sections IX-A4 and IX-B6 above. Modifying the controller per these suggestions should narrow the regulated pressure band. A third source of friction is the flexing of the diaphragms and of the backup discs. Since this flexing is beyond the elastic limit, the energy loss is not recoverable.

APPENDIX
Two Applications of the Generant Controllers

The generant controllers described in this Report were developed for use in systems in which a variable-pressure liquid supply must be metered to obtain a constant pressure in some downstream portion of the system.

One such system is the Advanced Liquid Propulsion System (ALPS) shown schematically in Fig. A-1 and Refs. 1 and 2. In this case the desired constant pressure is the propellant tank pressure. The propellant tank is pressurized by gases produced by decomposing hydrazine in a reactor called a gas generator. Hydrazine is supplied to the gas generator from a small auxiliary tank which operates on the "blowdown" principle (explained below); i.e., the hydrazine source pressure is variable. The flow of hydrazine from this variable pressure source is metered by the generant controller so that a constant pressure is maintained in the propellant tank.

In the ALPS, the rate of generant (hydrazine) flow is not necessarily constant. Variations occur for several reasons but primarily because the rate at which propellants are withdrawn from the main tank is modulated as the rocket engine which consumes these propellants is throttled by flow control valves in the injector. Thus, to keep a constant pressure in the propellant tank, the rate at which pressurizing gases are generated must be properly varied.

As mentioned above, this task of metering the generant flow is performed by the generant controller which, in this case, acts essentially as a remote sensing regulator. As in all mechanical pressure regulators, the force produced by the sensed pressure (propellant tank pressure) is compared with a reference force (produced by an adjustable spring). When these two forces do not balance, the controller alters the flow rate. Flow rate is increased if the sensing pressure force is low compared with the reference force, and it is decreased if the sensing pressure force is the greater. The detailed mechanics of this operation are described in the body of the report.

The flow rate through the generant controller must be independent of the source pressure because of the "blowdown" principle employed in the generant storage tank. To achieve proper blowdown operation, the small tank is partially filled with liquid and then the remaining volume is pressurized with an inert gas to a moderately high pressure. As the system operates and liquid is withdrawn, the pressure decays because of the expansion of the gas into the volume vacated by the liquid. Because no additional gas is added after initial pressurization, the pressure varies considerably; for the ratio of liquid volume to gas volume chosen for the ALPS, the maximum initial pressure of 1500 psi may be reduced to a minimum of only 500 psi as the last of the liquid is withdrawn.

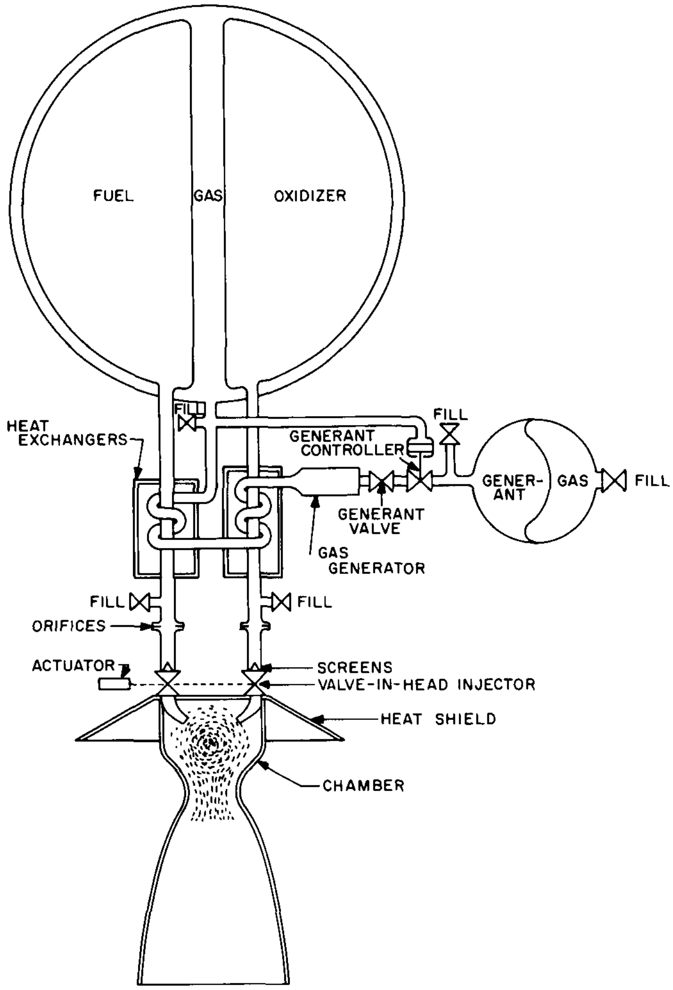

Fig. A-1. Schematic diagram of ALPS

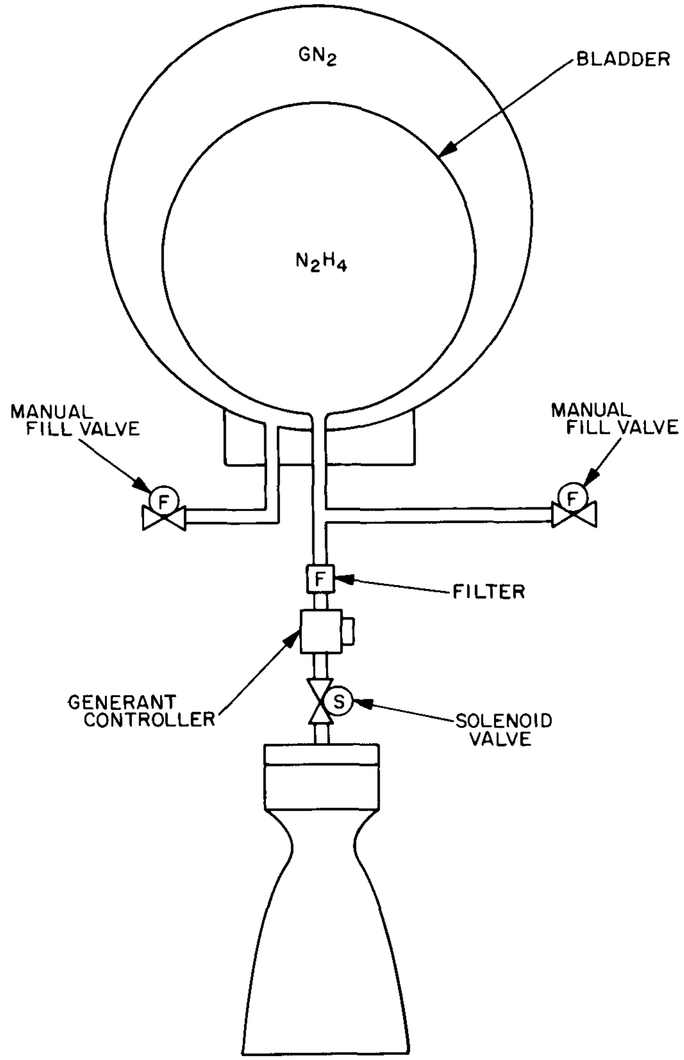

Fig. A-2. Schematic diagram of *Mariner* '66 system

At this writing, tests of the ALPS pressurization circuit are being made. Initial indications are that the generant controller performs as expected. Reports on these tests will appear in the JPL *Space Programs Summaries*, Vol. IV, No. 37-35.

This same blowdown operating principle has been applied to a small monopropellant rocket system which was tested as part of the *Mariner* '66 program. By using a circuit identical to the ALPS gas-generator feed circuit and a 50-lb thrust engine (from a Ranger spacecraft) packed with a spontaneously ignited hydrazine catalyst, it was possible to build a system which was considerably more simple than the conventional monopropellant rockets now flying in unmanned spacecraft. This application requires the propellant to be supplied to the engine at a constant flow rate and pressure so the engine will produce constant thrust.

In order for the generant controller to deliver a constant pressure to the engine from a decaying pressure source, the controller was installed with its sensing pressure port connected to its outlet port. This system is shown in Fig. A-2. With this arrangement the decline in source pressure has no effect on the pressure fed to the engine since the controller modulates the flow as necessary to keep the sensed pressure (outlet pressure in this instance) constant. Several test firings lasting up to 190 sec (limit of propellant supply) were made. In each case the pressure delivered to the engine during the steady-state operation varied no more than $\pm\frac{1}{2}$ psi. System operation was smooth and stable.

REFERENCES

1. Porter, R. N., and Evans, D. D., "Advanced Propulsion System for Unmanned Spacecraft," *Astronautics and Aeronautics Magazine*, June 1965.

2. Porter, R. N., Bond, D. L., Stanford, H. B., MacGlashan, W. F., and Keller, O. F., "Advanced Liquid Propulsion Systems," *Space Programs Summaries No. 37-32*, Vol. IV, Jet Propulsion Laboratory, Pasadena, California, April 1965, pp. 151–162.